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EFFICIENCY TESTS OF STEAM-TURBINE NOZZLES

By Prof. Frederick H. Sibley, University, Ala.
Member of the Society

T. S. Kemble,[1] Cleveland, O.
Non-Member

In 1905 a series of tests was begun at Case School of Applied Science, Cleveland, O., to determine the proper proportions and the efficiencies of steam-turbine nozzles for given steam conditions. The final tests, from which the results given herewith are derived, were made in an apparatus designed by T. S. Kemble, of the Chase Machine Co., of Cleveland. The writers spent about two years on these tests, and through the generosity of the company and the facilities afforded at Case School, were able to procure apparatus of considerable precision.

THEORY OF NOZZLES

2 By the theory of steam nozzles, a given weight of steam must pass all sections of the nozzle in the same time and the nozzle should be so constructed that the expansion will take place between the given initial and terminal pressures and wholly within the nozzle, the steam filling it completely at all sections. The available heat energy of the steam will then all be converted into kinetic energy and the efficiency will be a maximum. By efficiency we understand the kinetic energy of the jet per unit mass, divided by the available heat energy of the steam per unit mass.

3 To find the correct relation between the length and the cone angle of the nozzle, that the efficiency may be a maximum, is the problem to be determined experimentally. Let

[1] T. S. Kemble, Experimental Engineer for the Chase Machine Company, Cleveland, O.

All papers are subject to revision.

W = weight of steam flowing in pounds per second.
V = velocity of the jet in feet per second.
M = mass.
F = reaction in pounds.
g = acceleration due to gravity at Case School = 32.16015.
K = kinetic energy of the jet in foot-pounds or B.t.u.
E = total available heat energy of the steam.
Efficiency = $K \div E$.

4 From Mechanics we have: $K = \frac{1}{2}MV^2$ and $M = W \div g$. If $W = 1$ lb., then

$$K \text{ (the B.t.u.)} = \frac{V^2}{2g \times 778}$$

also

$$V = \frac{Fg}{W} \quad \text{for any flow}$$

F is the factor to be determined by experiment.

METHODS SUGGESTED

5 Three methods may be suggested for determining the efficiency of steam nozzles.

a By measuring the force of the jet when allowed to impinge on an external surface.
b By investigating the character of the jet with a search tube inserted axially in the nozzle.
c By measuring the reaction of the nozzle when a jet of steam is flowing through it.

6 The first method involves complications which tend to cast some doubt upon the results obtained. The force upon the external surface may be modified by the character of the surface, by eddying and steam friction, and by the distance traveled by the jet after leaving the nozzle and before it reaches the surface. When the surface used is a flat plate perpendicular to the axis of the jet, the force may even vary from a maximum to a negative value according to the relative location of the plate and nozzle.

7 The second method was tried in a series of experiments to find the pressure in the jet at various sections of the nozzle. A search tube was inserted axially in the nozzle and the relation between pres-

sure and flow was compared with the theoretical relation, as calculated and plotted from the steam tables. The first search tube used was of drawn copper $\frac{1}{16}$ in. in outside diameter and $\frac{1}{64}$ in. in inside diameter. One end was closed and the other connected to a mercury column. A $\frac{1}{64}$-in. hole was drilled through the search tube at right angles to its axis. The tube was so actuated as to bring the holes into any desired section of the nozzle, and the pressure shown by the mercury gage was recorded.

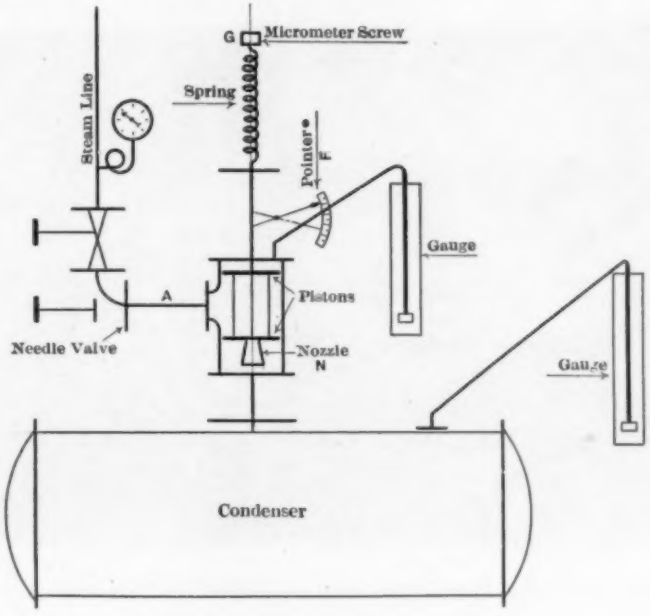


FIG. 1 DIAGRAM OF PISTON APPARATUS

8 This search tube was a failure, the pressures recorded being greatly at variance under identical conditions. This trouble was attributed to the capillary action on the condensed steam in the tube of the very small longitudinal hole.

9 A brass tube $\frac{3}{32}$ in. outside diameter and $\frac{1}{16}$ in. inside diameter gave results much more nearly consistent but not nearly accurate enough to determine the efficiencies at the various sections of the nozzle.

10 The third method provided for the determination of the reaction of the jet in the nozzle, and apparatus was constructed for this purpose, differing in detail as follows:

1 By fastening the nozzle into the outer face of one of a pair of rigidly connected pistons suspended in a cylinder. Steam entering between the pistons and flowing out through the nozzle would produce a measurable reaction.
2 By using a flexible steel tube suspended freely by one end and having the nozzle attached to a chamber at the other

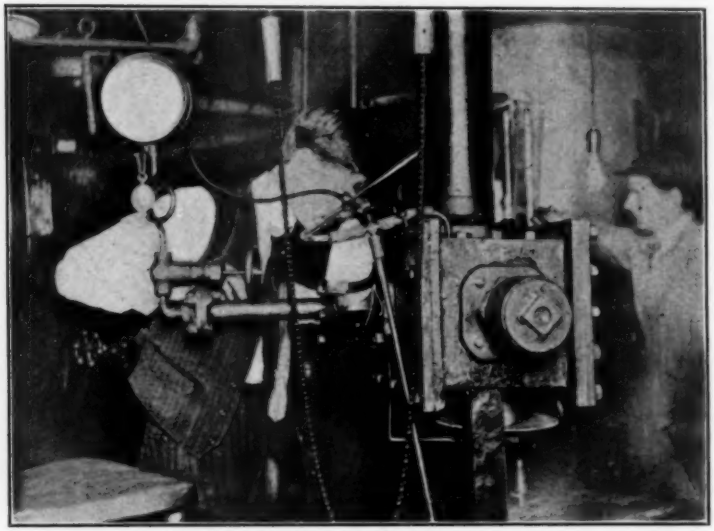


FIG. 2 FLEXIBLE TUBE APPARATUS FOR MEASURING REACTION OF JET

end with its axis perpendicular to the axis of the tube. Steam flowing downward through the tube and out of the nozzle would cause the tube to deflect with a measurable force.

PISTON METHOD

11 Fig. 1 is a diagram of the piston apparatus, having floating pistons rigidly connected with one another. High-pressure steam entered between the pistons at A and flowed out through the nozzle N to the condenser. The pistons were hung on the end of a calibrated spring enclosed within the exhaust steam space and their

weight put an initial tension in the spring. The total movement allowed the pistons was about $\frac{1}{16}$ in. and for convenience this movement was multiplied by the needle F in the ratio of ten to one. When steam flowed through the nozzle the reaction of the jet relieved part of the tension on the spring and the corresponding decrease of elongation was measured by the micrometer nut and screw and expressed in pounds.

12 As first constructed, the pistons moved quite freely when no steam was flowing, but when the steam was turned on the static fric-

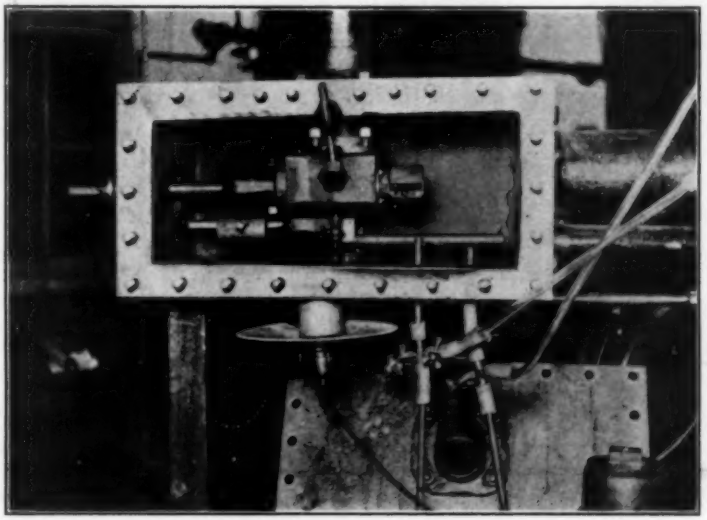


FIG. 3 INTERIOR OF BOX CONTAINING FLEXIBLE TUBE APPARATUS

tion increased so rapidly as to make accurate results impossible. When the diameter of the pistons had been reduced to the point where the friction became manageable, the leakage of steam past them became so great as to cause considerable trouble. Furthermore, condensed steam gathered on top of the upper piston, creating another source of error, because it was impossible to know the quantity present at any given instant and so to make a corresponding correction. Modification of this apparatus in the attempt to overcome these difficulties would have amounted almost to rebuilding and before doing this another method was tried with such success that it was not considered necessary to return to the old one.

FLEXIBLE TUBE APPARATUS

13 The flexible tube apparatus (shown in Figs. 2 to 5) was then tried with such success that it was not considered necessary to return to the piston apparatus. This apparatus was designed with a special view to combining the search-tube and reaction methods, by which means we hoped to obtain results more accurate than would result from either method alone.

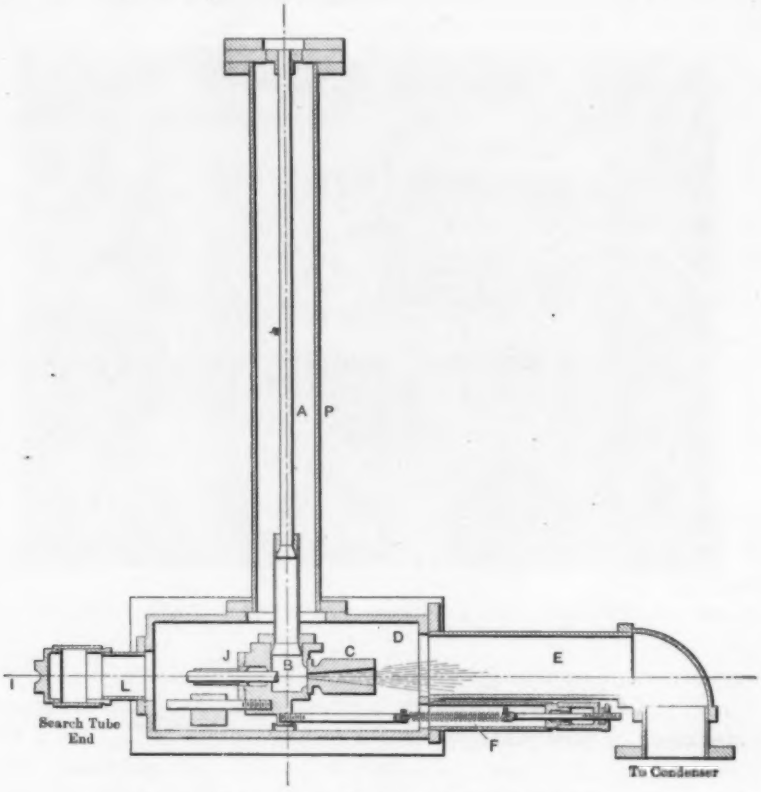


FIG. 4 SECTION OF FLEXIBLE TUBE APPARATUS

CONDITIONS OF TESTS

14 All tests were run at night, that there might be no interference with other work being conducted in the laboratory and boiler room. Steam was generated in a 125-h.p. Babcock & Wilcox boiler at a

pressure of 155 lb. gage with a variation of not more than 2 lb. up or down. There was about 50 deg. fahr. superheat at the boiler, which was connected with the testing apparatus by about 55 ft. of 5-in. pipe and 25 ft. of 4-in. pipe.

15 In the main steam pipe was located an angle needle-valve operated by a sprocket wheel and chain (Fig. 2) which made it possible to hold the nozzle feed-pressure very constant. From this point the steam passed downward through the tube *A* to the chamber *B*, (Fig. 4) thence through the nozzle *C* into the box *D*, and on through the passage *E* to the condenser.

16 The upper end of the tube *A* was screwed into a diaphragm on the lower flange of the angle valve. At its lower end it supported the chamber *B* which was allowed to move between stops restrained only by the stiffness of the tube and of the spring *F*. The motion of the chamber *B* was indicated by a needle which multiplied the motion

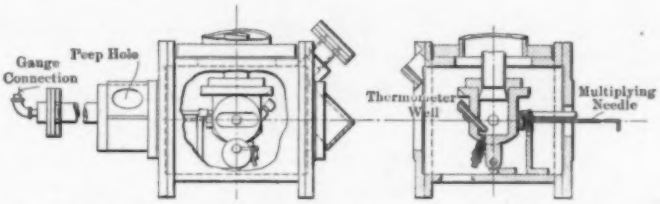


FIG. 5 END VIEWS OF SEARCH-TUBE END AND OF CONDENSER END

about ten to one. The spring *F* was operated by a micrometer nut and screw and was calibrated *in place* by known weights hanging on a flexible wire cable which extended from the back of the chamber in the line of the nozzle axis and down over a ball-bearing sheave. The tube *A*, the chamber *B* and the nozzle, were all enclosed in the vertical pipe *P* and the box *D*, and the vacuum surrounding them was greater than that in the condenser owing to the "augmenter" action of the steam jet entering the passage *E* to the condenser.

17 The initial temperature of the steam was shown by a thermometer inserted in a well in the chamber *B* and observed through a glass in the box D. A steam gage was connected by a flexible tube to the chamber *B*. The vacuum in the condenser and in the box *D* was shown by mercury columns, and the other column was joined by a flexible connection to a hole drilled as near as possible to the muzzle of the nozzle and perpendicular to the wall. All these connections to the mercury columns were of glass tubing with rubber couplings

which allowed the moving parts to swing freely without friction and made it easy to observe any accumulation of moisture, and by breaking the connection air could be let through to dry them quickly.

FORMS OF NOZZLE TESTED

18 The exact dimensions of the nozzles are shown in Fig. 6 and Fig. 7. They were all of machinery steel, bored taper, and had

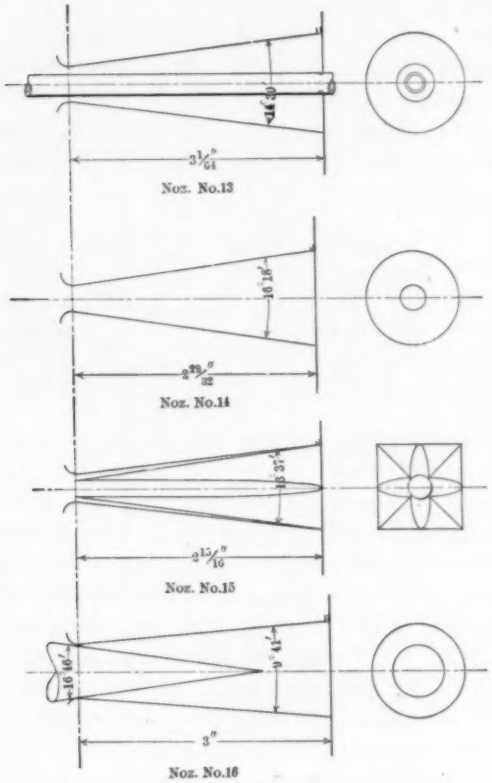


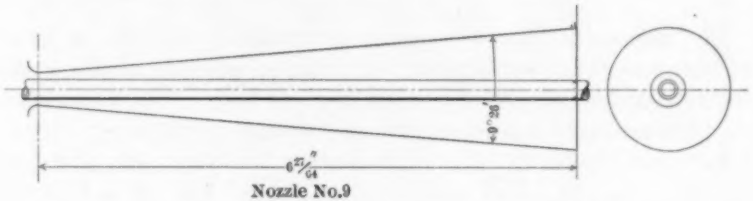
FIG. 6 FORMS OF NOZZLES TESTED

entrances rounded off with a hand tool to approximately $\frac{1}{8}$ in radius. All nozzles except No. 15 and No. 18 were bored smooth and polished. Nos. 10, 11 and 12 were identical except as to length and angle of divergence. No. 18 was like No. 11 except that it was finished rough

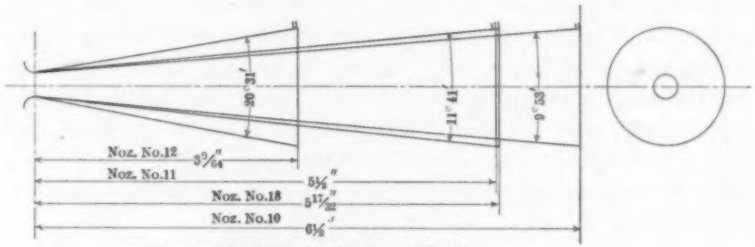
on the inside between the throat and muzzle, the finishing chip being made with a threading tool having an angle of 120 deg., and cutting 90 threads per inch. No. 14 and No. 15 were identical except that whereas No. 14 was bored taper, No. 15 was made in halves, and after

TABLE 1 DIMENSIONS OF NOZZLES IN FIG. 6

Nozzle No.	Gross Diameter Throat inches	Gross Diameter Outlet inches	Net Area Throat sq. ins.	Net Area Outlet sq. ins.	Length inches	Angle of Divergence
13	0.3949	1.156	0.0734	1.0005	$3\frac{1}{64}$	14° 30′
14	0.3038	1.128	0.0725	0.9993	$2\frac{31}{32}$	16° 18′
15	0.3038	1 in. sq.	0.0725	1.0000	$2\frac{1}{16}$	13° 37′
16	0.625	1.1315	0.0725	1.0055	3	9° 41′



Nozzle No.9



Nozzles No.10, No.11, No.12, and No.18

FIG. 7 FORMS OF NOZZLES TESTED

TABLE 2 DIMENSIONS OF NOZZLES IN FIG. 7

Nozzle No.	Gross Diameter Throat inches	Gross Diameter Outlet inches	Net Area Throat sq. ins.	Net Area Outlet sq. ins.	Length inches	Angle of Divergence
9	0.3940	1.4505	0.0728	1.6033	$6\frac{27}{64}$	9° 26′
10	0.3039	1.4240	0.0726	1.5926	$6\frac{1}{2}$	9° 53′
11	0.3039	1.4231	0.0725	1.5907	$5\frac{1}{4}$	11° 41′
12	0.3056	1.4241	0.0733	1.5930	$3\frac{9}{64}$	20° 31′
18	0.3039	1.4225	0.0725	1.5893	$5\frac{17}{32}$	11° 36′

being bored taper these halves were separated and milled longitudinally with a 90-deg. cutter, the cut beginning just beyond the throat and running deeper toward the muzzle, where the section becomes square, with the same area as the muzzle of No. 14. No. 16 was like No. 14 but had a larger throat area so that a needle point could be introduced to give the same net area as No. 14. No. 9 and No. 13 are search-tube nozzles made with throat and muzzle areas large enough so that the net areas, with the search tube in place, were equal to the net areas of the corresponding "reaction" nozzles. No. 9 corresponds to Nos. 10, 11, 12 and 18. No. 13 corresponds to Nos. 14, 15 and 16. The dimensions of the nozzles are given in Tables 1 and 2.

FLOW TESTS

19 Numerous tests were made to determine the rate of flow of steam through the various nozzles. Fig. 8 shows the results of these tests plotted to a scale of pounds flow per hour. The variations in flow are probably due principally to moisture in the steam, and to some extent to leakage from the water to the steam side of the condenser. The condenser was tested and at no time showed a leak exceeding two pounds per hour. There was sometimes a trace of superheat at the nozzle entrance, and this increased with an increase in the volume of flow. At pressures of less than 145 lb., moisture was probably always present. For this reason the values used in the calculations were the mean flow-values for 145-lb. pressure, and a trifle less than the mean for the lower pressures. It is to be regretted that we were unable to procure a calorimeter of sufficient accuracy for our purpose.

20 In Fig. 8 the results are given in pounds per hour for the four initial pressures. Each small circle represents the result of one flow test of from 15 min. to 30 min. duration. The vertical dotted lines represent the flow values that were used in the efficiency calculations. The flow values for the irregularly shaped nozzles are a little higher than the others, as is shown in the upper part of the diagram. The diagonal lines simply connect together the results found in the same test. For example, the five circles along the lowest line of the chart represent the values found for nozzle No. 9 on January 17 and 18, 1908. The vertical scale is not important although each initial pressure is located higher up on the diagram than the preceding one as a matter of convenience.

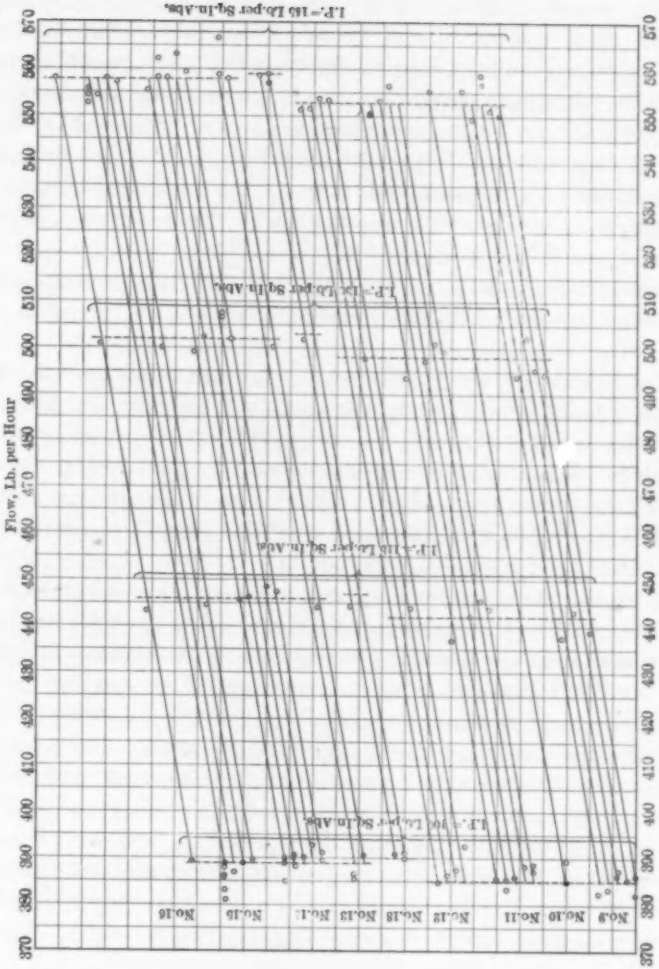


FIG. 8 WEIGHT OF STEAM FLOWING THROUGH NOZZLES ON DIFFERENT DATES

being bored taper these halves were separated and milled longitudinally with a 90-deg. cutter, the cut beginning just beyond the throat and running deeper toward the muzzle, where the section becomes square, with the same area as the muzzle of No. 14. No. 16 was like No. 14 but had a larger throat area so that a needle point could be introduced to give the same net area as No. 14. No. 9 and No. 13 are search-tube nozzles made with throat and muzzle areas large enough so that the net areas, with the search tube in place, were equal to the net areas of the corresponding "reaction" nozzles. No. 9 corresponds to Nos. 10, 11, 12 and 18. No. 13 corresponds to Nos. 14, 15 and 16. The dimensions of the nozzles are given in Tables 1 and 2.

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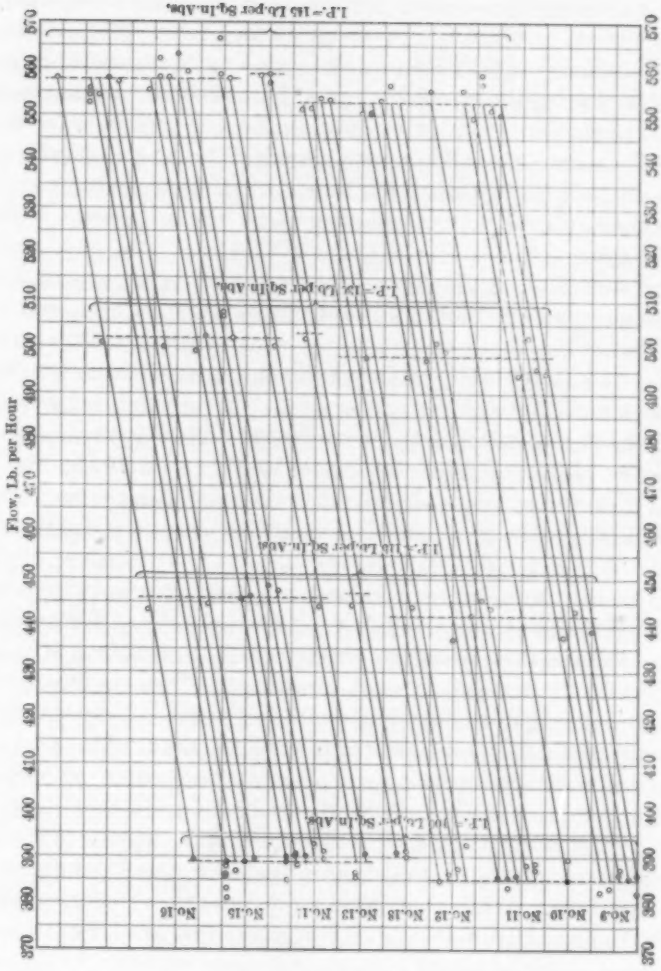

FIG. 8 WEIGHT OF STEAM FLOWING THROUGH NOZZLES ON DIFFERENT DATES

SPRING CALIBRATIONS

21 The accuracy with which the reactions were determined depended largely on the care taken in calibrating the springs. These calibrations were first made with the spring in place in the apparatus, and at the room temperature of about 85 deg. fahr. The readings were taken while gradually loading the spring to 25 lb. and then unloading it, repeating the operation a great number of times and taking average extension under any given load as the true extension for that load. The average extension under a load of 25 lb. was 1.4875 in., or 0.0595 in. per lb. In cooling off, tube *A*, Fig. 4, seemed to warp a little, so that after about two minutes there was a decrease in the initial extension of the spring of about 0.0025 in. As it took about two minutes to shut off the steam and get the initial extension after each reaction reading, the spring extensions were all corrected by this amount.

22 Another factor which affected the spring calibration was the change in temperature of the spring itself. A thermometer was inserted in the spring casing and the spring calibrated at various temperatures by observing the temperature and extensions simultaneously. From these temperature calibrations a correction factor $l[0.0002423\ (t_2 - t_1)]$ was obtained and used to correct the reactions found by using the factor 0.0595 in. per lb.

23 After the thermometer had been inserted in the spring casing it was noted that as the reaction test progressed the temperature at first increased and then remained nearly constant regardless of moderate changes in initial and terminal pressures. This was due to the fact that the box *D* stood open and cooled off between tests and then warmed up gradually when the steam was turned on. The average temperature in the spring casing was about 135 deg. fahr., or 50 deg. higher than the room temperature at which the original calibration was made. All the reactions found before this thermometer was in place were corrected on the assumption that the spring temperature was 135 deg. fahr. While this did not eliminate the error due to the fact that the temperature increased during the first part of each test, it did bring the average pretty close to what it should be.

24 Corresponding readings of spring extensions at the beginning and end of tests were lower and higher respectively than the average extension. This was due to the above-mentioned difference in spring temperature. When the spring temperature was read simultaneously with the spring extension this difference disappeared.

SEARCH-TUBE TESTS

25 After completing the flow tests and the spring calibrations one other factor remained to be determined before the reaction tests could be made. This was the determination of the pressure at the muzzle of the nozzle.

26 *The reaction of any nozzle is equal to the summation of all the components, parallel to its axis, of the pressures within the nozzle and in the chamber from which it leads.* If the pressure of the medium surrounding the nozzle and the chamber is equal to that in the plane of the muzzle, then the reaction as shown by the pull on the spring is the true reaction. If the pressure of the surrounding medium is greater than that in the plane of the muzzle it will decrease the apparent reaction, and if the pressure of the surrounding medium is less than that in the plane of the muzzle it will increase the apparent reaction. The amount of such increase or decrease will be equal to the difference in the unit pressure multiplied by the area of the muzzle. The true reaction of the nozzle is equal to the pull of the spring plus or minus this pressure difference.

27 The demonstration of this proposition possibly differentiates these experiments from those heretofore published, as the writers do not know of any other case where the combination has been used in this manner.

28 The muzzle pressure was found by using the search tube with nozzles No. 9 and No. 13. The search tube here used was a selected piece of cold-drawn Shelby tube $\frac{1}{4}$ in. in outside diameter and $\frac{3}{16}$ in. in inside diameter, with six holes $\frac{1}{32}$ in. in diameter drilled perpendicular to the axis. The outside of the tube was polished to micrometer measurement. The chamber B was rigidly connected to the back wall of the box D by the distance piece at J (Fig. 3). The rear end of the search tube was encased and supported by a tube at L which had on its outer surface a thread fitted with a micrometer nut, and passed through the distance piece holding the search tube in the axis of the nozzle.

29 The holes in the search tube were located in the same plane as the hole in the wall of the nozzle. One gage was connected to the box D, another to the hole in the wall of the nozzle, and a third to the rear end of the search tube. Simultaneous readings of these gages were taken with varying pressures in the box. These readings were plotted in Fig. 9 and Fig. 10.

30 The diagonal lines represent the box pressures, which were

varied from 0.5 lb. to 2 lb. absolute. The larger circles represent the pressures at the rim of the nozzle and the smaller circles that at the center of the nozzle. These are plotted for the four initial pressures as found on different dates. The dotted horizontal lines represent the muzzle pressures used in determining the true reaction. For example, on May 22 and 23, with an initial pressure of 100 lb. under a box pressure of 0.8 lb. absolute the terminal pressure at both rim and center of the nozzle was 0.7 lb. absolute; 0.648 was the average terminal pressure used in the calculations.

31 The first tests seemed to indicate a higher pressure in the center of the stream than in the rim. Later a leak in the search-tube connections was discovered and repaired, and the tests repeated. This time the rim readings were constant, while the search-tube readings came down toward the rim readings, showing that they had been affected by the leak. More careful repairs still further reduced the leak and the search-tube readings, until there appeared to be no actual difference in pressure at the rim and at the center of the stream. It was decided to use the rim readings where a difference remained as it was hardly possible to keep the search-tube connections perfectly tight throughout a night's run, and error seemed more likely than in the rim readings.

32 Fig. 9 and Fig. 10 show that the pressure in the center of the stream remains constant with considerable variation in pressure in the box D, and the pressure at the rim of the stream remains constant until the pressure in the box rises somewhat above that shown in the search tube, at which time the pressure works gradually in along the rim without at first affecting the center of the stream.

33 The terminal pressures taken from these tests were further corrected by the difference between the pressure in this plane and in the exact plane of the muzzle. This correction involved the subtraction of only 0.01 lb. per sq. in. for No. 9, and 0.03 lb. per sq. in. for No. 13.

REACTION TESTS

34 In making reaction tests the following routine was observed: The barometer was read with every observation, and corresponding corrections made so that the initial absolute pressure used should be exactly 145, 130, 115 and 100 lb. per sq. in. This was for convenience in making computations. One observer maintained a constant initial pressure by manipulating the needle valve in the steam pipe. A second observer maintained a constant vacuum in the box D by

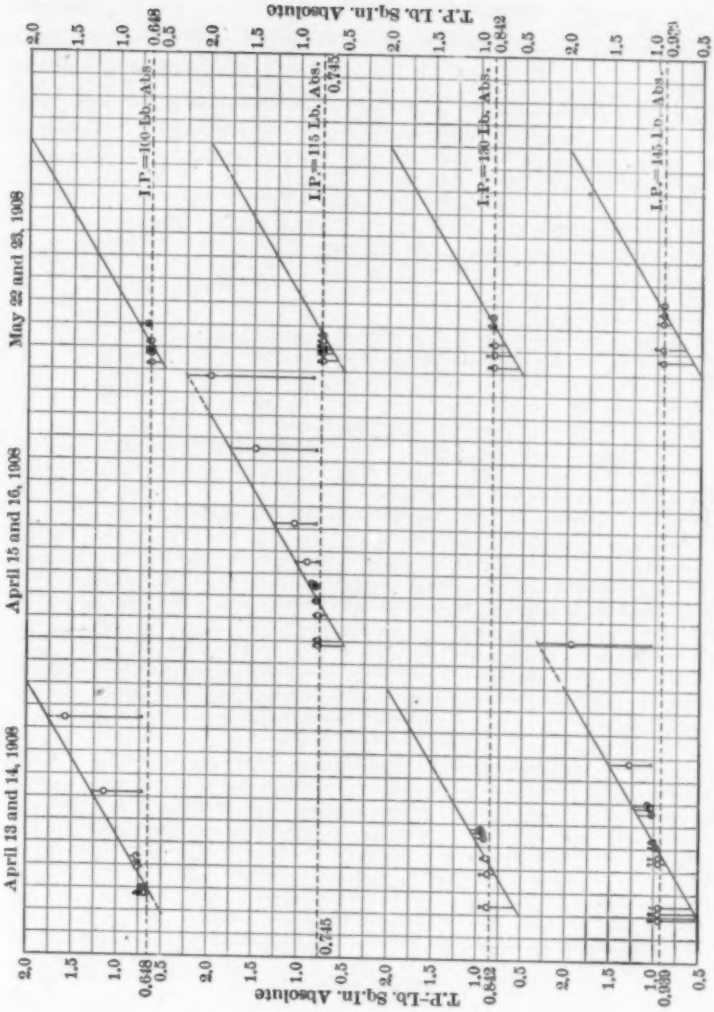

FIG. 9 EFFECT UPON THERMAL PRESSURE OF NOZZLE, PRODUCED BY VARYING PRESSURE IN BOX IN WHICH NOZZLE WAS SUSPENDED

THIS IS FOR NOZZLE NO. 9, WHICH CORRESPONDS TO NO. 11.

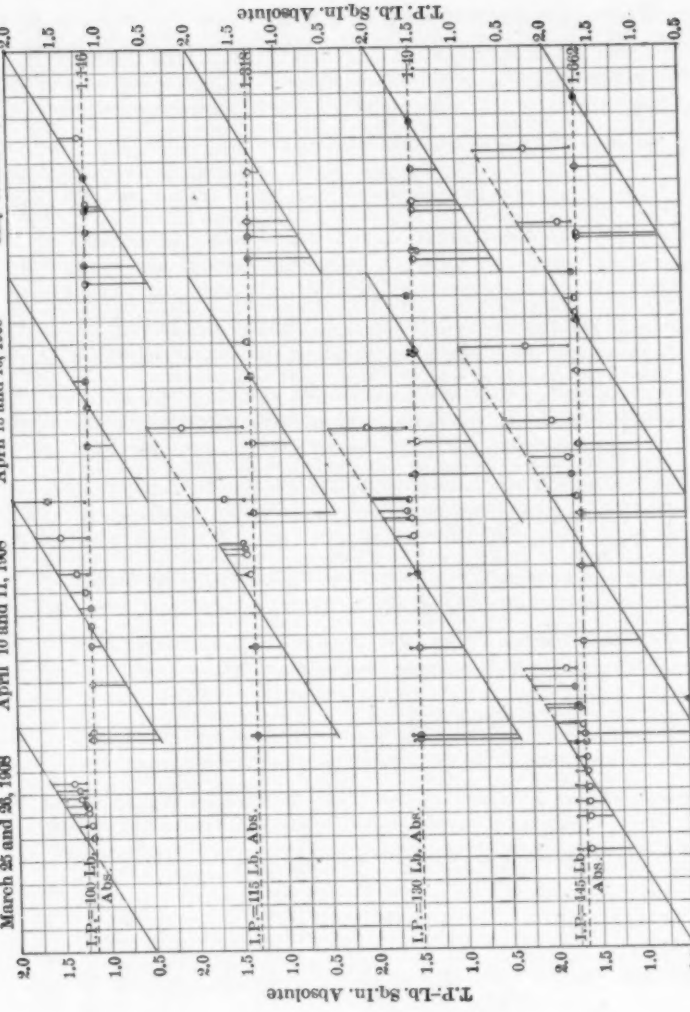

FIG. 10 EFFECT ON THERMAL PRESSURE OF NOZZLE, PRODUCED BY VARYING PRESSURE IN BOX IN WHICH NOZZLE WAS SUSPENDED

THIS IS FOR NOZZLE NO. 13, WHICH CORRESPONDS TO NO. 14.

manipulating the valves leading to the pump and condenser. A third operated the micrometer screw which registered the spring extension and thus held the multiplying-needle opposite an index at the center of its travel. A fourth and sometimes a fifth man read gages, and one man was generally occupied in moving about behind the observers to check observations.

35 As the reaction of the jet forced the chamber B back against

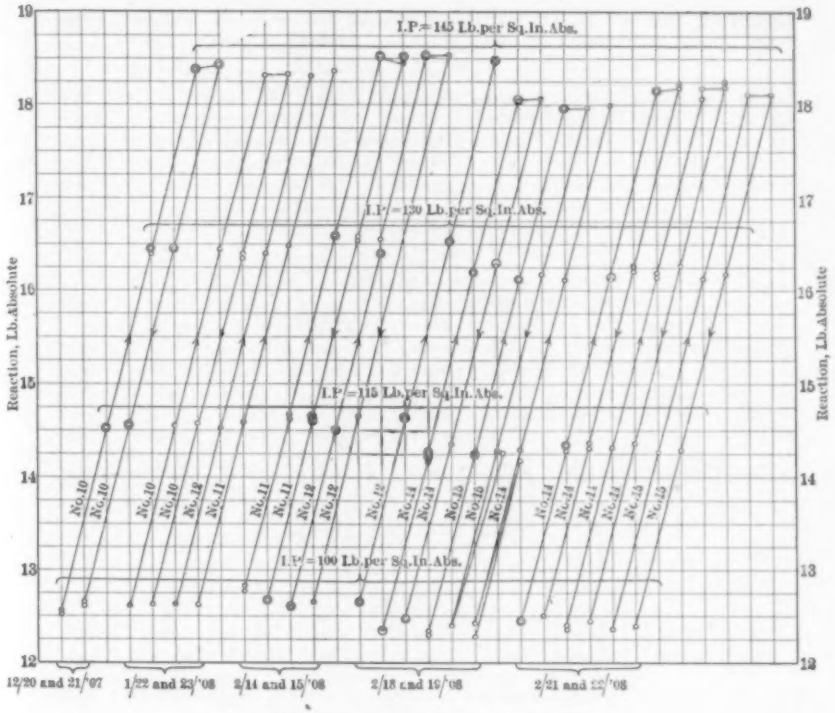

FIG. 11 PRELIMINARY REACTION TESTS

the tension of the spring, the micrometer screw was worked forward until the tension in the spring balanced the reaction and the multiplying-needle indicated that B was swinging freely in the central position. When the needle remained quiet for an instant in this position the observer gave a signal and the thermometer and gages were read simultaneously. After each reading the steam was shut off, and the position of the micrometer screw with the needle in the central position was noted. The difference between this position and

that when the steam was flowing gave the elongation of the spring due to the reaction.

36 It will be noted that friction has been entirely eliminated in this apparatus, except for the trifling amount due to the movement of the multiplying-needle.

CALCULATION FOR EFFICIENCY

37 A series of preliminary reaction tests was run on all of the nozzles with the result that nozzles No. 11 and No. 14 were selected as

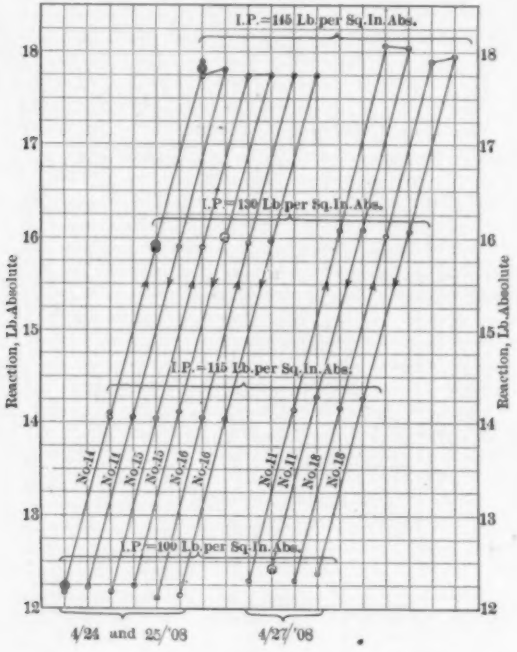

FIG. 12 COMPARISON OF TESTS ON NOZZLES 14, 15 AND 16 WITH THOSE ON NOZZLES 11 AND 18

representative for future tests. The results of the preliminary tests upon these nozzles are given in Fig. 11. Exhaustive tests of long duration were then made upon nozzles No. 11 and No. 14 with the results shown in Fig. 13 and Fig. 14, and in Tables 3 and 4. Fig. 12 shows the comparative results of certain of the tests.

38 In Fig. 11 and Fig. 12 the reactions in pounds absolute are grouped together horizontally. The reactions are shown for each

of the four initial pressures. The vertical scale is four lines to the pound. The horizontal scale is not important, the initial pressures being placed diagonally over each other as a matter of convenience. The diagonal lines connect the reactions of a single nozzle at the various pressures. The arrowheads indicate whether the reactions

TABLE 3 COMPUTATION OF EFFICIENCY, NOZZLE NO. 11

I. P. Abs.	T. P. Lbs. per Sq. In. Abs.	Reaction	Flow Lbs. per Hr.	Flow Lbs. per Sec.	Vel.	B.T.U.$_1$	B.T.U.$_2$ Table	Eff. %
145	0.929	18.134	553	0.1536	3796	288.0	317.4	90.75
145	1.029	17.974	553	0.1536	3763	283.0	312.5	90.55
145	0.829	18.294	553	0.1536	3830	293.1	322.7	90.84
145	0.929	18.134	558	0.1550	3762	282.9	317.4	89.13
145	0.929	18.134	548	0.1522	3831	293.3	317.4	92.41
145	0.929	18.234	553	0.1536	3817	291.2	317.4	91.75
145	0.929	18.034	553	0.1536	3776	284.9	317.4	89.75
130	0.832	16.244	498	0.1383	3776	285.0	315.4	90.36
115	0.735	14.351	442	0.1228	3759	282.4	313.6	90.03
100	0.638	12.45	385	0.1069	3744	280.1	311.5	89.91
100	0.738	12.29	385	0.1069	3696	273.0	304.6	89.63
100	0.538	12.61	385	0.1069	3792	287.4	319.5	89.94
100	0.638	12.45	390	0.1083	3696	273.0	311.5	87.62
100	0.638	12.45	380	0.1056	3793	287.5	311.5	92.29
100	0.638	12.55	385	0.1069	3774	284.6	311.5	91.36
100	0.638	12.35	385	0.1069	3714	275.6	311.5	88.47
			Assuming 2 per cent moisture					
145	0.929	18.134	553	0.1536	3796	288.0	311.9	92.35
130	0.832	16.244	498	0.1383	3776	285.0	310.0	91.93
115	0.735	14.351	442	0.1228	3759	282.4	308.2	91.62
100	0.638	12.45	385	0.1069	3744	280.1	306.1	91.51

B.t.u. are given only to the nearest tenth, and for this reason efficiencies are not accurate in second decimal place.

B.t.u.$_1$ = equivalent of kinetic energy of jet in B.t.u.

B.t.u.$_2$ = available heat energy of steam.

shown on that line were taken when the pressure was increasing or decreasing. The circles represent the actual reactions in pounds as plotted from the tests. Two circles occurring together indicate that two independent readings of the reaction were taken at the same time. A larger circle outside the small one indicates the reading preferred.

39 Fig. 13 shows graphically the result of a complete series of reaction tests on nozzle No. 11, which is practically the same as nozzles Nos. 9 and 13. The vertical scale is 20 lines to the pound, while the horizontal scale is 100 lines to the pound. The full diagonal lines connect together the observed reactions in pounds under a varying box pressure. The horizontal dotted lines connect together

TABLE 4 COMPUTATION OF EFFICIENCY, NOZZLE NO. 14
SHOWING EFFECT OF ERROR IN DETERMINATION OF TERMINAL PRESSURE FLOW OR REACTION

I. P. Abs.	T. P. Lbs. per Sq. In. Abs.	Reaction	Flow Lbs. per Hr.	Flow Lbs. per Sec.	Vel.	B.T.U.$_1$	B.T.U.$_2$ Table	Eff. %
145	1.632	17.821	558	0.1550	3698	273.2	289.8	94.28
145	1.732	17.721	558	0.1550	3677	270.2	286.7	94.23
145	1.532	17.921	558	0.1550	3718	276.3	293.0	94.30
145	1.632	17.821	563	0.1564	3665	268.4	289.8	92.61
145	1.632	17.821	553	0.1536	3731	278.2	289.8	95.99
145	1.632	17.921	558	0.1550	3718	276.3	289.8	95.34
145	1.632	17.721	558	0.1550	3677	270.2	289.8	93.22
130	1.46	15.977	502	0.1394	3685	271.3	288.2	94.15
115	1.288	14.147	446	0.1239	3672	269.5	286.5	94.07
100	1.116	12.295	389	0.1081	3659	267.6	284.7	93.99
100	1.216	12.195	389	0.1081	3630	263.3	280.3	93.92
100	1.016	12.395	389	0.1081	3689	272.0	289.3	94.01
100	1.116	12.295	394	0.1094	3613	260.8	284.7	91.62
100	1.116	12.295	384	0.1067	3707	274.6	284.7	96.45
100	1.116	12.395	389	0.1081	3689	272.0	284.7	95.53
100	1.116	12.195	389	0.1081	3630	263.3	284.7	92.47
				Assuming 2 per cent moisture				
145	1.632	17.821	558	0.1550	3698	273.2	284.7	95.97
130	1.46	15.977	502	0.1394	3685	271.3	283.1	95.84
115	1.288	14.147	446	0.1239	3672	269.5	281.5	95.74
100	1.116	12.295	389	0.1081	3659	267.6	279.7	95.67

B.t.u. are given only to the nearest tenth, and for this reason efficiencies are not accurate in second decimal place.

B.t.u.$_1$ = equivalent of kinetic energy of jet in B.t.u.

B.t.u.$_2$ = available heat energy of steam.

the same reactions after being corrected for the difference in pressure between the box and the muzzle of the nozzle. The dotted diagonal line represents the pressure in the muzzle of the nozzle as found with the search tube and is plotted to the same horizontal scale as the box pressure. When the box and terminal pressures are the same the apparent and corrected reactions are the same, as

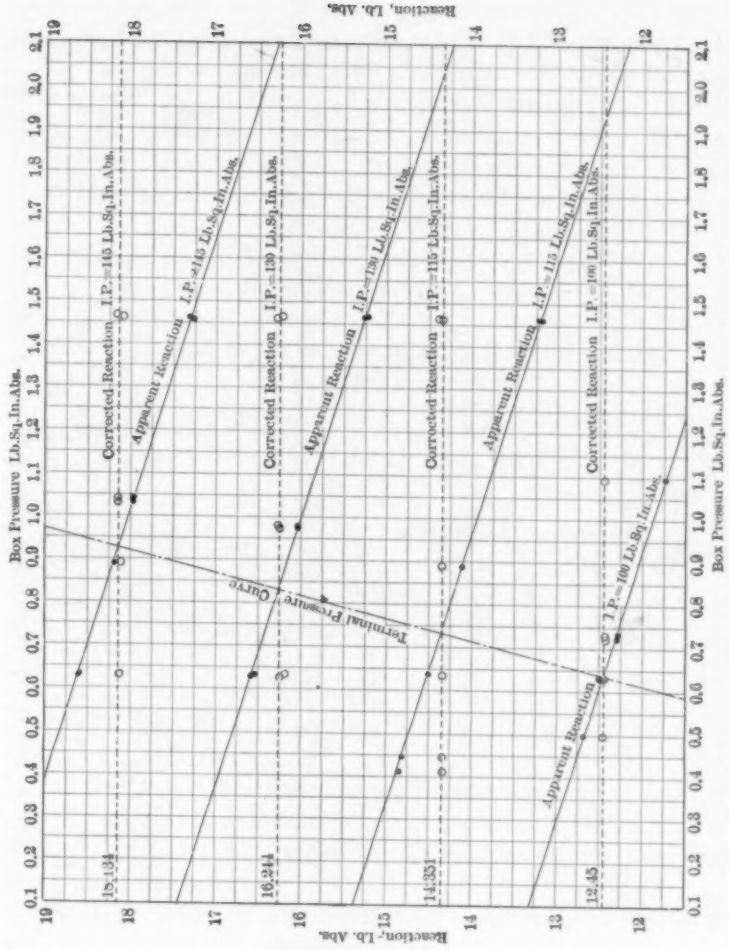

FIG. 13 REACTION TESTS OF NOZZLE NO. 11 WITH VARYING BOX PRESSURE

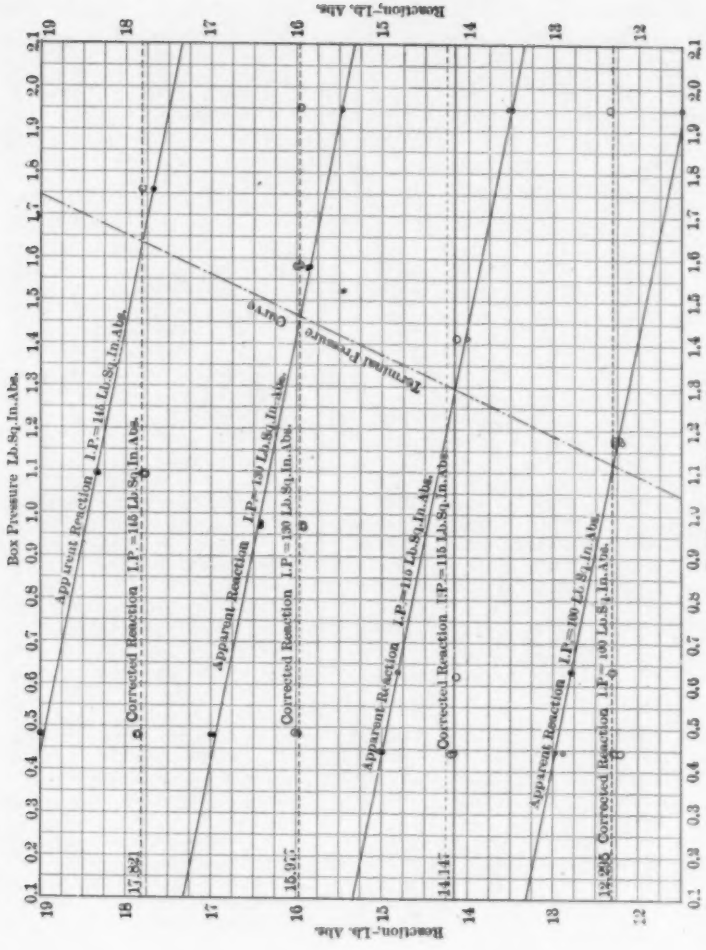

FIG. 14 REACTION TESTS ON NOZZLE NO. 14 WITH VARYING BOX PRESSURE

is shown by the intersection of the three lines. As the box pressure increases above that point where the terminal and box pressures were the same the apparent reaction falls below the true reaction, and as the box pressure falls below that point where the terminal and box pressures were the same, the apparent reaction rises above the true reaction. As an example, the terminal pressure at 115 lb. initial pressure is approximately 0.7341 lb. When the box pressure was increased to 0.89 the apparent reaction was 14.10 lb. To this apparent reaction was added the correction factor (difference in pressure $\times$ area of muzzle) to give the corrected reaction 0.14.351. Fig. 14 is like Fig. 13, the values being for nozzle No. 14.

40 The method of calculation may be illustrated by the following from nozzle No. 14 with an absolute initial pressure of 145 lb. per sq. in. The terminal pressure (See Fig. 10) is 1.662 lb. per sq. in., and deducting 0.03 this becomes 1.632 lb. per sq. in. The reaction (Table 1) is 17.821 lb. The flow is 558 lb. per hr. or 0.155 lb. per sec.

$$\text{Velocity} = V = \frac{\text{reaction} \times g}{\text{flow (lb. per sec.)}} = \frac{17.821 \times 32.16}{0.155} = 3697.6 \text{ ft. per sec.}$$

$$\text{B.t.u.}_1 = \text{kinetic energy of jet} = \frac{V^2}{2g \times 778} = \frac{3697.6^2}{2 \times 32.16 \times 778} = 273.22$$

$$\text{B.t.u.}_2 = \text{available energy (from steam table)} = 289.8$$

$$\text{Efficiency} = \frac{\text{B.t.u.}_1}{\text{B.t.u.}_2} = \frac{273.22}{289.8} = 0.9428 \text{ or } 94.28 \text{ per cent}$$

41 If the terminal pressure had been determined as 1.732 and no other factor changed, the true reaction as calculated would have been 17.721 instead of that shown in Table 3 and the resulting efficiency would be 94.23 per cent.

42 If the flow had been determined as 563 lb. per hr. with no other change of values the efficiency would have figured 92.61 per cent.

43 A reaction of 17.921 without other change would have given an efficiency of 95.34 per cent.

44 If we assumed 2 per cent of moisture, the efficiency would figure as 95.97 per cent.

45 The efficiency was also calculated by the search-tube method, by first plotting curves (similar to those shown in Fig. 15) showin

the relation between pressure and rate of flow per unit area of section, with adiabatic expansion and with various percentages of friction loss. The pressure and flow found by experiment were then plotted on this chart and the efficiency determined graphically by comparison.

46 The chart in Fig. 15 was designed for finding the nozzle efficiency by the search tube method. The vertical scale is four lines to the pound. The horizontal scale is 10 lb. per hour per line. The chart shows the relation between the flow in pounds per square inch of section and the pressure in any section of the nozzle, assum-

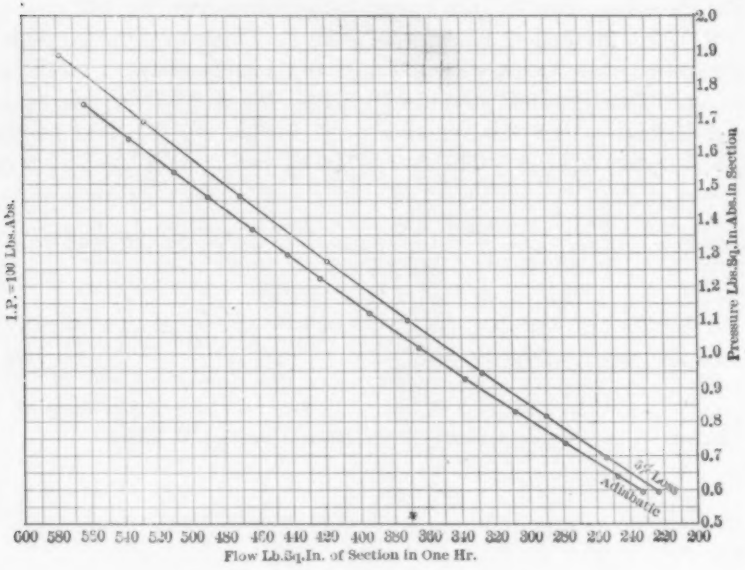

FIG. 15 PRESSURE-FLOW CHART

ing adiabatic expansion for the lower curve and 5 per cent loss of heat for the upper one. The data for plotting these curves was obtained from the steam tables. By plotting the observed values of pressure and flow upon these sheets we are able to obtain a graphic solution for efficiency. For example, in the case of nozzle No. 9 the terminal pressure at 100 lb. initial pressure was found to be 0.638 lb. per sq. in. and the corresponding flow in pound per square inch of section is 238 ±. From the chart, assuming adiabatic expansion, under terminal pressure of 0.638 the flow would be 248 lb. and 238/248 is approximately 96 = the efficiency.

TABLE 5 SUMMARY OF CALCULATIONS OF EFFICIENCY FOR NOZZLES 9, 11, 13 AND 14

Nozzle No.	I. P. Lbs. per Sq. In. Abs.	T. P. Lbs. per Sq. In. Abs.	Flow Lbs. per Hour	Reaction Pounds	Theoret. B.t.u. per Lb.	Calcul. B.t.u. per Lb.	Theoret. Vel. Ft. per Sec.	Calcul. Vel. Ft. per Sec.	Eff. %
9	100	0.638	385	.....	.....	.....	.....	.....	96.1
	115	0.735	442	.....	.....	.....	.....	.....	96.3
	130	0.832	498	.....	.....	.....	.....	.....	96.0
	145	0.929	553	.....	.....	.....	.....	.....	95.5
11	100	0.638	385	12.45	311.5	280.1	3948	3744	89.9
	115	0.735	442	14.351	313.6	282.4	3962	3759	90.0
	130	0.832	498	16.244	315.4	285.0	3973	3776	90.4
	145	0.929	553	18.134	317.4	288.0	3985	3796	90.7
13	100	1.116	389	.....	.....	.....	.....	.....	98.3
	115	1.288	446	.....	.....	.....	.....	.....	97.9
	130	1.46	502	.....	.....	.....	.....	.....	97.5
	145	1.632	558	.....	.....	.....	.....	.....	97.1
14	100	1.116	389	12.295	284.7	267.6	3774	3659	94.0
	115	1.288	446	14.147	286.5	269.5	3796	3672	94.1
	130	1.46	502	15.977	288.2	271.3	3798	3685	94.2
	145	1.632	558	17.821	289.8	273.2	3808	3698	94.3

TABLE 6 EFFECT OF ERROR IN OBSERVATIONS

Nozzle No.	Error in Observation		Corresponding % Error in Efficiency	
	+	−	+	−
9	0.1 lb. per sq. in.	Terminal Pressure		8.5 to 14.0
11	0.1 lb. per sq. in.	Terminal Pressure		0.03 to 0.3
13	0.1 lb. per sq. in.	Terminal Pressure		5.4 to 9.2
14	0.1 lb. per sq. in.	Terminal Pressure		0.02 to 0.07
9	5 lb. per hr.	Flow	1.0 to 1.6	
11	5 lb. per hr.	Flow		1.6 to 2.4
13	5 lb. per hr.	Flow	1.1 to 1.5	
14	5 lb. per hr.	Flow		1.67 to 2.46
9	2%	Dryness Factor	0.9 to 0.6	
11	2%	Dryness Factor		1.6
13	2%	Dryness Factor	0.9 to 0.7	
14	2%	Dryness Factor		1.7
11	0.1 lb.	Reaction	1.0 to 1.45	
14	0.1 lb	Reaction		1.05 to 1.54

Note.—An error of +0.1 lb. would be caused in the calculated reaction by an error of +0.1 lb. per sq. in. in the box pressure-reading of No. 14 or by an error of + 0.0628 lb. per sq. in. in the box pressure-reading of No. 11.

RESULTS AND CONCLUSIONS

47 Table 5 gives a summary of the calculations for nozzles No. 9, 11, 13 and 14, and Table 6 a summary showing what would be the effect of error in the observed values. The discrepancy between the efficiencies calculated for the search-tube and the reaction nozzles is principally due to the great difference in the effect on the two methods of calculation of a small error in terminal pressure. An increase of only $\frac{1}{20}$ lb. per sq. in. over the value used would cover the discrepancy. Corrections for any slight condenser leak which may have existed would decrease the flow values and bring the calculated efficiencies closer together.

48 The terminal pressures chosen were the minimum observed values. The dryness factor was assumed as 100 per cent. Assuming a 2 per cent moisture would make the calculated efficiencies for No. 13 and No. 14 very nearly equal, and very materially reduce the difference between No. 9 and No. 11.

49 In consideration of the above, taken in connection with Tables 2 and 3, we may assume that the values 91.5 per cent for No. 9 and No. 11, and 95 per cent for No. 13 and No. 14, are probably within 2 per cent of the true efficiencies. No. 15 and No. 16 show a trifle less reaction than No. 14 but the flow also appears to be a trifle less, and there is not sufficient ground for assuming any difference between the efficiencies of these three nozzles. Neither is there any appreciable difference in Nos. 10, 11 and 12. No. 18, with a greater flow and less reaction than No. 11, shows an efficiency of about three per cent less. Since no appreciable difference in efficiency is shown either with a variation in cone angle from 9 deg. to 20 deg., or with the variations in contour shown in nozzles No. 15 and No. 16, smoothness of finish would appear to be a much more important factor than contour.

GOVERNING ROLLING MILL ENGINES

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Associate Member of the Society

In considering the conservation of steam-power equipment for driving rolling mills, we must take into account the two methods of rolling: the two-high mill driven by a reversing engine, and the three-high mill driven continuously in one direction; and the relative amount of power required for each.

2 There is very little variation in the type used for each class of mill. Twin engines are used for two-high mills and single engines for three-high mills, usually tandem compounds.

3 The reversing engine for the two-high mill must be powerful enough to take care of the engine and mill friction and the maximum torque produced by the piece in the rolls in any position. As these engines are usually twin engines with cranks at 90 deg., each side must be capable of doing the work alone when the other side is on the dead center.

4 In determining the size and distribution of the metal in engines of this type it is the custom to make the dimensions a little larger and the parts a little heavier than have been used before for the same work. Reciprocating parts are made heavier to stand the shocks, thereby increasing their inertia, and making necessary heavier frames, bedplates, bearings and pins, as well as more rigid adjustments, which in turn require more attention.

5 As an example of the power sometimes used for an engine of this type, a certain engine may be cited which was fully capable of delivering 25,000 h.p. while the actual average work on the steel passing through the mill could not have required more than 2000 h.p. at the maximum capacity of the mill. The engine and mill friction, if the mill were driven continually in one direction, would not fall much short of 1000 h.p. Assuming that each reversal would absorb 50 h.p., and that there were ten reversals per minute, 500 h.p. would be con-

All papers are subject to revision.

sumed in this way. The total average work of the engine was thus about 3500 h.p., only one-seventh of its capacity.

6 As the three-high mill is driven continually in one direction, the energy stored in the flywheel makes it possible to do the same work with considerably less than one-half the maximum power required in the former case, the amount depending upon the size and weight of the flywheel. The greater the amount of energy the wheel can store up, the closer can the maximum power required approach the average work of the mill, resulting in the more economical use of steam and a lower cost of equipment.

7 Mill designers do not always give sufficient consideration to this fact and operators have to deal later with high steam cost and difficulty in keeping the proper steam pressure. Of course, there are other features to be considered, but economical use of power is a very important item.

8 For driving a three-high mill a twin engine of the cross-compound type could be used with an intercepting valve such as is employed in locomotive practice, by which the engine could be started from any position and handled by a quick-acting throttle valve, so that it could be brought to a standstill as soon as a piece passed through the rolls, if another were not ready to enter the mill. Such an arrangement would go a long way toward answering one of the principal arguments in favor of the two-high mill: that its engine uses steam only when the piece is on the mill.

9 If an engine of the type described be furnished with a very heavy flywheel located between the engine and the mill, the shocks due to the piece striking the rolls will be taken very largely by the flywheel and not by the engine. Furthermore, if the engine were so designed that it could not work through the wide range of steam admission, as is the current practice, the abnormal amount of compression now necessary would be cut down to a large extent, the parts of the engine would be strained less, and the engine would run with greater steam economy owing to the cutting out of the high release pressure during heavy work and the reduction of the number of strokes during the period of negative work.

10 Going into the details of operation of an engine driving a three-high mill, we find:

a That the engine first develops just enough power to take care of the friction of engine and mill.

11 If the engine were running at a constant speed during this

period, just enough steam would be admitted to the cylinder at each revolution to do the work with the least possible variation of cut-off, resulting in the most economical use of the steam. The more the speed varies the greater the amount of steam required per horsepower developed.

12 A constant speed is also desirable for another reason; namely, the available energy stored in the flywheel is always normal under these conditions, whereas with a varying speed it would be below the normal about one-half the time. Should the piece strike the rolls when the steam pressure is low and the steel cold, the engine would be more liable to stall if the stored energy were below normal than if it remained constant.

b Next, the piece strikes the rolls.

13 The initial force of this blow is absorbed by the flywheel and the speed of the engine is reduced in consequence. When this has dropped, say four or five revolutions, the governor has probably so adjusted the steam valves that the engine is developing its maximum power and the valves will remain in this adjustment until the engine is nearly up to speed again. During this time the release pressure will be high, making it necessary to carry a very high compression, conditions under which a non-condensing engine will make the most noise. Further, if the steam going to waste were utilized the engine would be capable of doing about one-third more work.

c Next, the piece leaves the rolls.

14 On many passes the engine is receiving the maximum amount of steam at this instant and the flywheel absorbs energy through the increase of speed above normal. In the writer's opinion, this is the time when nearly all of the failures of flywheels on rolling mill engines occur, and anything that can be done to cut down the amount of energy to be absorbed by the wheel at this time will increase the safety of the engine and decrease the repairs on the engine and mill.

15 Having in most cases reached the highest point of speed, the governor will shut off steam entirely from the cylinder, the engine will slow down to several revolutions below normal speed before sufficient steam will be admitted to increase this speed again and the result will be a wide range of the speed variation during the first period when the engine has simply to overcome its own friction and that of the mill.

16 The operations outlined in the foregoing are very complicated when two or more passes occur at the same time.

17 The results of a study of the above details of operation of the No. 1 rail mill engine driving the four-pass roughing rolls at the Ensley Rail Mill of the Tennessee Coal Iron and Railroad Company, during a series of tests show how great a difference in the performance of the engine the writer was able to obtain with one adjusting screw added to the governor.

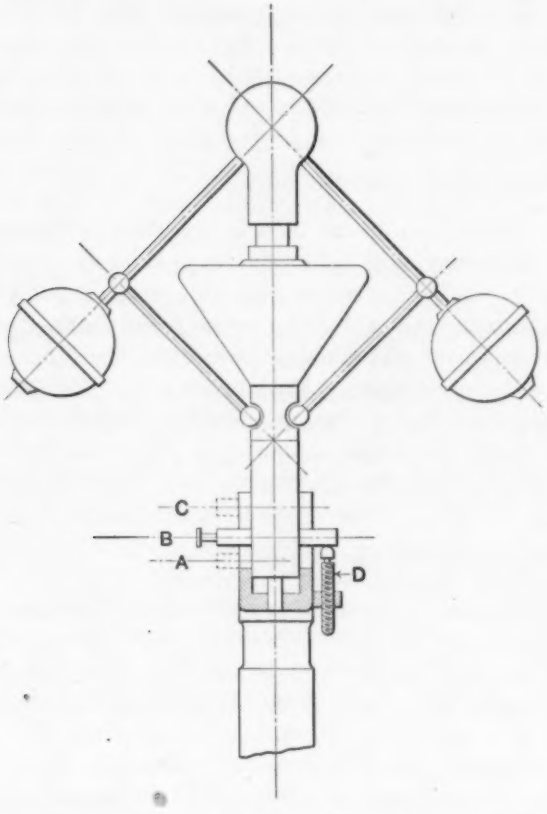

FIG. 1 SHOWING LOCATION AND FUNCTION OF THE ADJUSTING SCREW *D* ON THE GOVERNOR

18 The engine is a Reynolds Corliss, 52 in. by 72 in., non-condensing, equipped with a long-range cut-off valve gear.

19 Fig. 1 shows the location of the adjusting screw *D* on the governor. Its purpose is to prevent the governor from dropping to position *A* which would allow the maximum amount of steam to

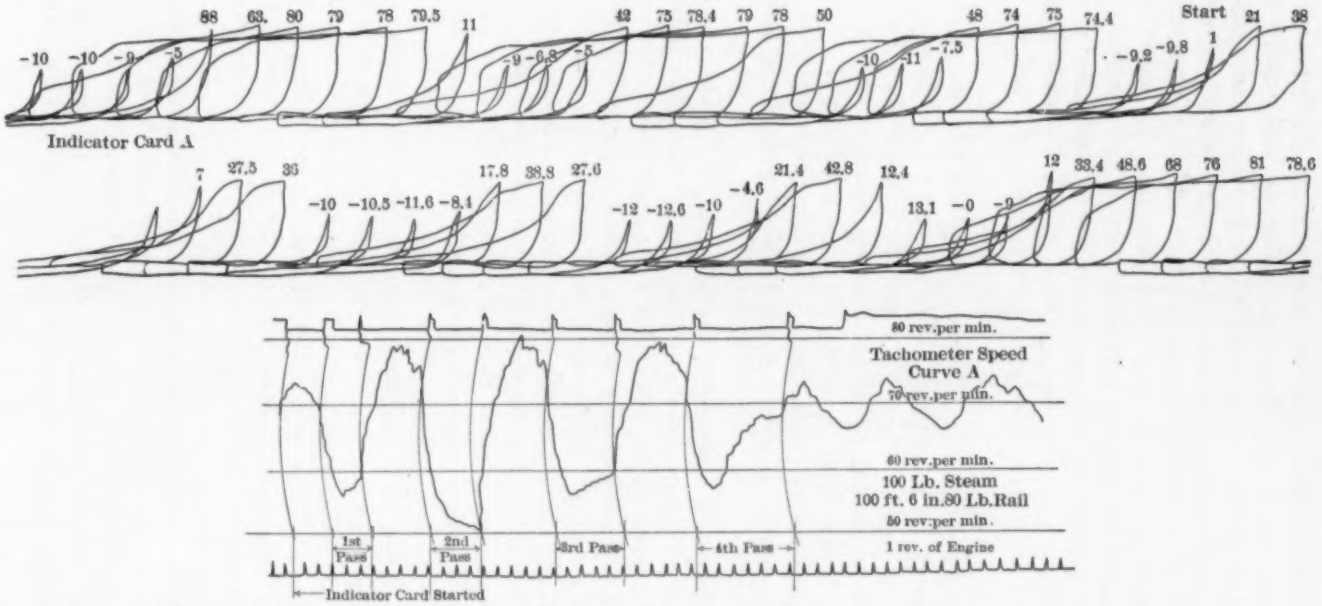

FIG. 2 CONTINUOUS INDICATOR CARD *A* AND TACHOMETER SPEED CURVE *A* OF NO. 1 RAIL MILL ENGINE, 52 IN. BY 72 IN.

THE CARD AND CURVE WERE TAKEN AT THE SAME TIME AND SHOW ORDINARY PERFORMANCE. THE LOWER HALF OF INDICATOR CARD IS A CONTINUATION OF THE UPPER HALF

reach the cylinder, as determined by the valve gear. In this case the steam was admitted nearly three-quarters of the stroke with the governor in this position. When the pin is in position *C* the steam is entirely cut off. The screw *D* was adjusted very slowly while the engine was under load to determine the most advisable position *B*.

20 This position must be at a point such that the engine will carry an average load and also will not stop under a heavy load. It can readily be seen that by reducing the range of adjustment of steam distribution the engine will operate with more economical steam consumption and the greatest strains on the engine and mill will be reduced.

21 The degree of success attained by this adjustment may be judged somewhat by the accompanying continuous indicator cards and tachometer speed curves. Card and curve marked *A* (Fig. 2) were taken together before the attachment was used. Card and curve *B* (Fig. 3) were taken when the attachment was in use.

22 The speed curves were made by a recording tachometer which the writer rigged up from a Schaeffer & Budenberg indicating tachometer.

23 This indicating instrument was mounted on a plate, a bevel gear being fastened to the end of its driving shaft, which in turn drives a shaft with a worm at one end. The worm shaft is mounted on a bracket which will swing out of gear, thus disconnecting the indicating instrument if desirable. The worm wheel driven by the worm shaft runs loose on one of the paper-feed rolls and when a record is to be taken a small clutch causes the worm wheel to drive the rolls. Two rolls are geared together and a third acts as a press roll. The paper rolls used were such as are furnished for the Uehling pneumatic pyrometer.

24 The indicating needle on the original tachometer was replaced by a longer one which reached to the paper. A pencil was attached to the end of the new needle but there was so much friction that the records were of no value, so a small tin funnel was fastened to the needle and a linen thread passed through the hole in the bottom, protruding about $\frac{1}{8}$ in. The ink in the funnel worked down the thread to the paper and made a satisfactory record.

25 Below the paper-supporting plate is a vertical plate to which are fastened two electric bells with the gongs removed and pencils substituted for the clappers. One of these bells was operated by a contact made once in a revolution of the engine, the record being shown at the bottom of the curve *A* (Fig. 2.) As it was very evident

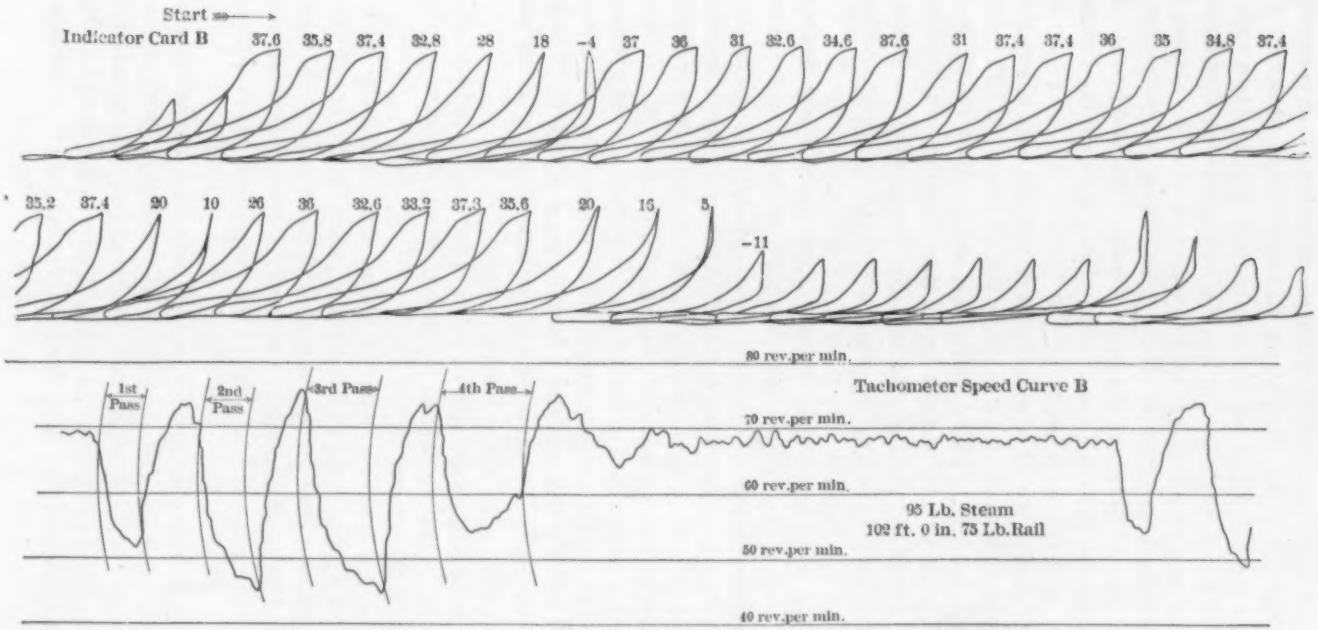
Start
Indicator Card B
37.6 35.8 37.4 32.8 28 18 -4 37 36 31 32.6 34.6 37.6 31 37.4 37.4 36 35 34.8 37.4
35.2 37.4 20 10 26 36 32.6 33.2 37.3 35.6 20 15 5
-11
80 rev.per min.
1st Pass
2nd Pass
3rd Pass
4th Pass
Tachometer Speed Curve B
70 rev.per min.
60 rev.per min.
95 Lb. Steam
102 ft. 0 in. 75 Lb. Rail
50 rev.per min.
40 rev.per min.

FIG. 3 CONTINUOUS INDICATOR CARD *B* AND TACHOMETER SPEED CURVE *B* TAKEN WHEN RUNNING WITH THE CAINE METHOD OF ENGINE CONTROL

THE LOWER HALF OF THE INDICATOR CARD IS A CONTINUATION OF THE UPPER HALF

that the paper would always feed the same amount at each revolution it was not considered necessary to use this device each time.

26 The other bell was operated on a circuit that had two gaps in series, one of which was closed when the indicator cards were started and the other kept closed except at the instant an observer at the rolls indicated the start and stop of the various passes, by momentarily breaking the circuit by means of a push button. This last feature also appears unnecessary, as the speed curves have a pronounced change of direction at these instants.

27 From card *A* it will be noted that during the four passes there are 37 records showing that the engine is taking steam, and 22 records showing that it is not. These would lead one to think that if every card were a positive one, and the work were distributed throughout the entire period between pieces, and the mean effective pressure averaged, there would be a more economical use of steam, and possibly with a heavy flywheel the size of cylinder could be reduced. That would be the ideal condition, which cannot be realized, however, because of three changing functions: the varying time between pieces, the varying temperature of the steel and the varying steam pressure.

28 Card *B*, Fig. 3, shows the engine doing the same work as before, on a piece of the same length, and it can be seen that the work is distributed over 34 revolutions and only two negative cards. With the engine running as this card shows, it is a comparatively easy proposition to set the valves for economical steam distribution and it is also much easier to keep the rods and boxes properly adjusted. The low terminal pressure is the cause of the engine's running much more quietly than when card *A* was taken.

29 The indicated steam consumption of card *A* is about 43 lb. of steam per h.p. per hr., against 37 lb. in card *B*, a saving during rolling periods of over 20 per cent.

30 Of the speed curves, curve *A* shows that during the friction load the engine varies from 66 to 73 r.p.m., with an average of about 69 revolutions, and after the passes the speed becomes about 80 r.p.m., an increase of 11 revolutions above normal. Curve *A* shows also that the second pass is the heaviest one of the four, the speed dropping to 51 r.p.m.

31 Curve *B* indicates that during friction load the speed varies only about 3 r.p.m., and that the highest velocity is 75 r.p.m., or 7 above normal. Some changes were made on the rolls between the two records so that the third pass was as heavy as the second and the speed drops to about 45 r.p.m. These curves indicate that the engine

TABLE OF DATA FOR SUCCESSIVE PASSES

No. 1 Rail Mill Engine, E. P. Allis, 52 in. × 72 in. 4 Roughing Passes, Rolls 27½ in. Pitch Diameter, Product 100 ft. 6 in., 80-lb. Rail, Total Weight, 2680 lb.

		First Pass	Intervals	Second Pass	Intervals	Third Pass	Intervals	Fourth Pass	Intervals	Total
1	Area section after pass, sq. in.	54		33.5		22.9		19		
2	Length section after pass	15.95		25.7		37.6		45.2		
3	Revolutions	2.23	4.77	3.59	5.41	5.25	5.25	6.31	4.19	37
4	Speed at start } from curve	70.0	58	76	51	76.5	59	74	68	
5	Speed at finish } from curve	58.0	76	51	76.5	59.0	74	68	69	
6	Total m.e.p., pass and intervals	461.7		702.6		719.5		677.2		
7	Total energy pass and intervals, ft. lbs.*	5,755,000		8,758,000		8,968,000		8,440,000		
8	Total energy, ft. lbs.*	2,602,200	3,152,800	4,573,500	4,184,500	6,276,300	2,691,700	8,008,800	431,200	
9	Energy taken from flywheel, 1180 $(n^2_1 - n^2_2)$	1,812,500		3,747,000		2,798,000		1,005,000		
10	Energy added to flywheel		2,846,000		3,836,500		2,354,000		161,700	
11	Energy, total	4,414,700		8,320,500		9,074,300		9,013,800		
12	Energy in friction, 64,320 ft. lb. per rev	143,400	306,800	230,900	348,000	337,700	337,700	406,000	269,500	2,380,000
13	Energy, total, less friction in piece	4,271,300		8,089,600		8,736,600		8,607,800		29,705,300
14	Torque per foot length of piece	268,000		315,000		232,000		191,000		

*From cylinders 12,464 × m.e.p.

could be speeded up to about 75 r.p.m. and not exceed the speeds used before, that more energy was stored in the flywheel and that the engine would not drop below the speed shown on Curve *A*.

32 Referring to the table, the constant used in items 7 and 8 is the foot-pounds of work for one stroke of the engine at 1 lb. mean effective pressure.

33 In the original calculation for item 9 only the flywheel was considered. This is 22 ft. in diameter, weighs about 130,000 lb. and has a radius of gyration of about 7.4 ft. Taking the formula

$$\text{energy} = \frac{Wv^2}{2g}$$

and altering it to get the energy stored up or given out at a change in velocity of one revolution per minute it becomes

$$E \text{ for 1 r.p.m.} = \frac{\text{wt. wheel} \left(\frac{\text{radius gyration} \times 2\pi}{60 \text{ seconds}} \right)^2}{64.32}$$

Substituting and solving with the values given above

$$E \text{ for 1 r.p.m.} = 1220 \text{ ft.-lb.}$$

As the energy varies as the square of the velocity we would use the following to represent the amount of energy involved in a change of velocity

$$E = 1220 \; n_1^2 - n_2^2$$

in which n_1 is the higher number of revolutions of the engine and n_2 the lower number. In checking this up against indicator card and speed curve results on friction alone, it became evident that 1180 was the proper constant to use to include the inertia effects of the mill and reciprocating parts.

34 The constant in item 12 was the average foot-pounds of work per revolution during the friction period as calculated from the indicator cards. The slide rule was used in making the calculations.

COOLING TOWERS FOR STEAM AND GAS-POWER PLANTS

WITH PARTICULAR REFERENCE TO THE POSSIBILITIES OF THE NATURAL-DRAFT AND AUXILIARY-DRAFT TYPE

By J. R. Bibbins, New York

Member of the Society

The object of this paper is to bring to the attention of the members of the Society a subject which has received relatively little attention in the past, but which the author believes merits the careful study of all engineers interested in future power-plant development. The cooling tower has been looked upon as a makeshift, and its use has been correspondingly restricted. This, however, is largely due to the extremely limited information of an exact or technical nature, relative to the possible performance under unfavorable weather conditions, available to the general public, as well as to the somewhat uncertain factor of depreciation. And, further, it is the author's belief that the present high prices[1] constitute the greatest obstacle to the more widespread adoption of the cooling tower in both turbine and gas-power plants.

2 Believing that interest may be aroused in this subject by a more widespread dissemination of engineering data, the author will present for discussion a type of tower with which some personal experience has been acquired, and suggest a type of combined fan and natural draft which would be suited to most efficient running on peak as well as light loads. It is not the intention to discredit the cooling tower in its present forms but rather to bring about a more general recognition of its inherent advantages.

[1] Some recent quotations from a number of builders of forced-draft towers suitable for a load of several thousand kilowatts, (*not including* the motor or engine for driving the fan), ranged between $4.80 and $6.93 per kw., as much as the entire condensing equipment.

All Papers are subject to revision.

PRESENT FIELD

3 There is a continual demand for cooling towers from inland power stations where the condensing water supply is costly or restricted. Turbine-driven plants, as a rule, operate with higher vacuum than those engine-driven, with the result that the perform-

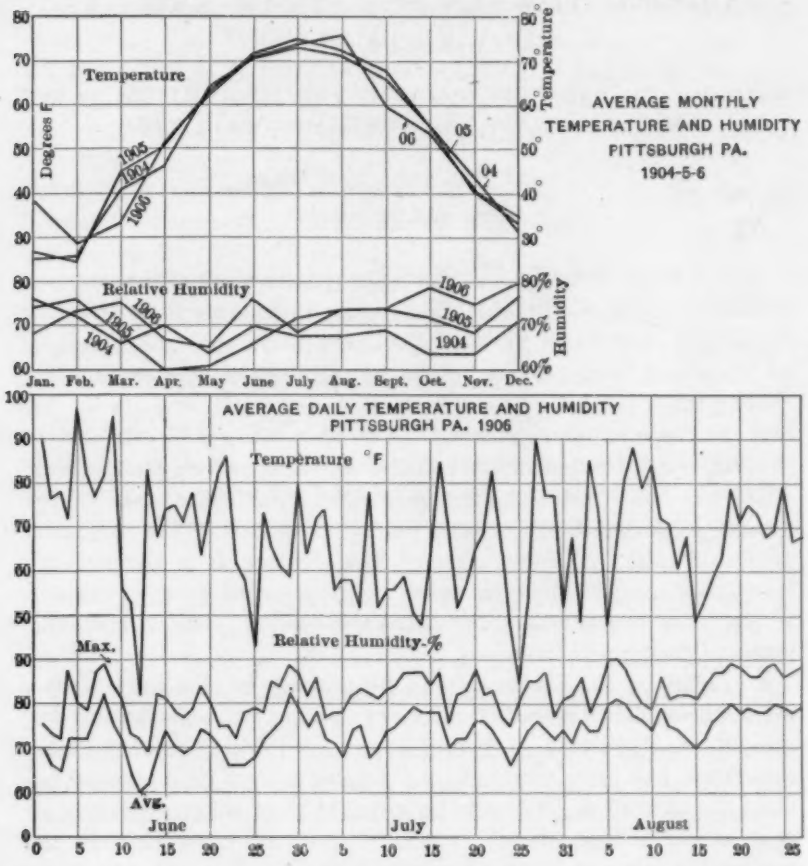

FIG. 1 CYCLES OF AVERAGE TEMPERATURE AND HUMIDITY AT PITTSBURGH, PA.

ance demanded of the tower must be proportionately better. In the past, cooling towers have generally been associated with badly run plants and low vacuum. This, however, is clearly a question of design and adaptation and not an inherent fault. To be sure, service requirements are not easily met (see Fig. 1). Not infrequently atmos-

pheric temperatures of 90 deg. to 100 deg. fahr. are encountered, with cooling water at 80 deg. to 90 deg. fahr. and humidity above 85 per cent saturation. Yet the auxiliary plant must be moderate in bulk and the power consumption low. Furthermore, it must be capable of overload capacity to tide over daily peak loads and periods of exces-

TABLE 1 WEATHER CONDITIONS, PITTSBURGH, PA.

DATE	AVERAGE TEMPERATURE DEGREES FAHR.	AVERAGE HUMIDITY PER CENT
1904	52.8	73
1905	52.1	68
1906	52.0	70
Average.....	52.3	70.5

MAMIXUM TEMPERATURE AND HUMIDITY RANGES, SUMMER OF 1906

TEMPERATURE ABOVE	DAYS IN MONTH			AVERAGE FOR MONTH
	JUNE	JULY	AUGUST	
90 deg.	..	..	2	0.6
85	6	9	14	10.5
80	8	14	10	20.5
75	10	6	4	27.7
70	5	2	1	23.7
Below	..	..	..	..

HUMIDITY ABOVE	DAYS IN MONTH			AVERAGE FOR MONTH
	JUNE	JULY	AUGUST	
90 %	3	1	3	2.3
80	14	10	15	13.0
70	8	11	11	10.0
60	3	8	2	4.3
50	2	1	..	1.0
Below	..	..	..	..

Data from Pittsburgh Weather Bureau.

sively hot and humid weather, and all of this with small investment cost.

4 Curiously enough, there is an active demand for cooling towers in the South, e. g., Florida, where the atmospheric conditions are the most unsuitable; also in the Western mining and coast regions. For-

tunately, low humidity prevails here, as a general rule (e. g., Colorado Springs ranges around 50 per cent).

5 In power-gas work, the demand for cooling towers is especially pressing. The large quantity of water required for engine jackets and for gas cooling and washing, entails a heavy expense if water is scarce and costly. Gas-engine discharge water, being quite pure, should not be wasted, but cooled and returned to the plant. Even with deep well pumps supplying sufficient water for cooling, a station

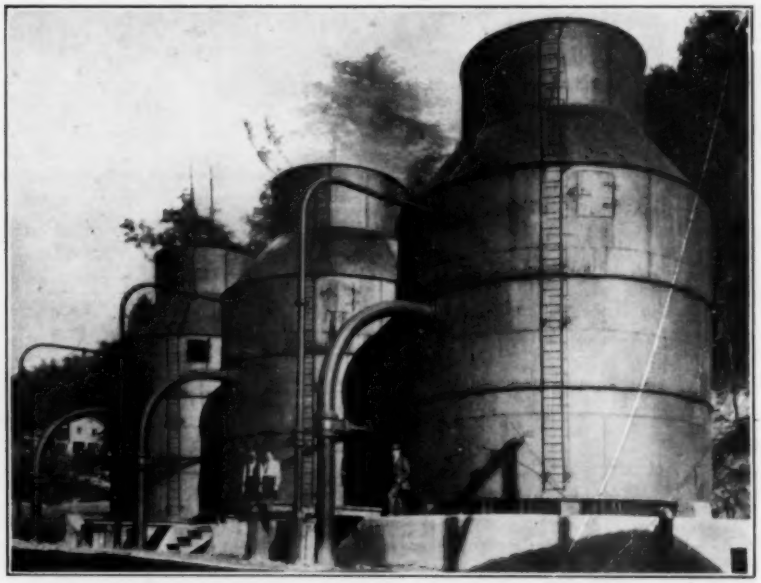

FIG. 2 4500-H.P. NATURAL-FORCED-DRAFT COOLING TOWER AT GARY, W. VA., FOR A LOW-PRESSURE AND HIGH-PRESSURE TURBINE INSTALLATION

THESE TOWERS HAVE AUXILIARY FANS IN THE STACK DRIVEN BY PELTON WHEELS AND SMALL TURBINE-DRIVEN ROTARY PUMPS LOCATED IN THE POWER STATION, AND OPERATE WITH NATURAL DRAFT WHEN NOT HEAVILY LOADED

is handicapped by a large expense for auxiliary power consumption. In one instance of an Arizona mining plant, the only water available for cooling was so impure as to make it necessary to install a completely *closed cooling system* for the engine jackets, in which no evaporation took place, simply cooling by conduction. In city light and power plants, not fortunate enough to be located on water frontage, cooling towers built upon the roof have been utilized for engine cool-

FIG. 3 1500 H.P. OPEN-TYPE STEEL TOWER AT SAN LUIS POTOSI, MEXICO, FOR A GAS-POWER CENTRAL STATION

ONE TOWER SERVES THE ENGINE JACKETS, THE OTHER THE PRODUCER SCRUBBER. THE SIDES ARE ENCASED BY WIRE SCREEN TO REDUCE THE WATER LOST BY WINDAGE. COOLING, 10 TO 30 DEG. HUMIDITY GENERALLY LOW, AROUND 50 PER CENT.

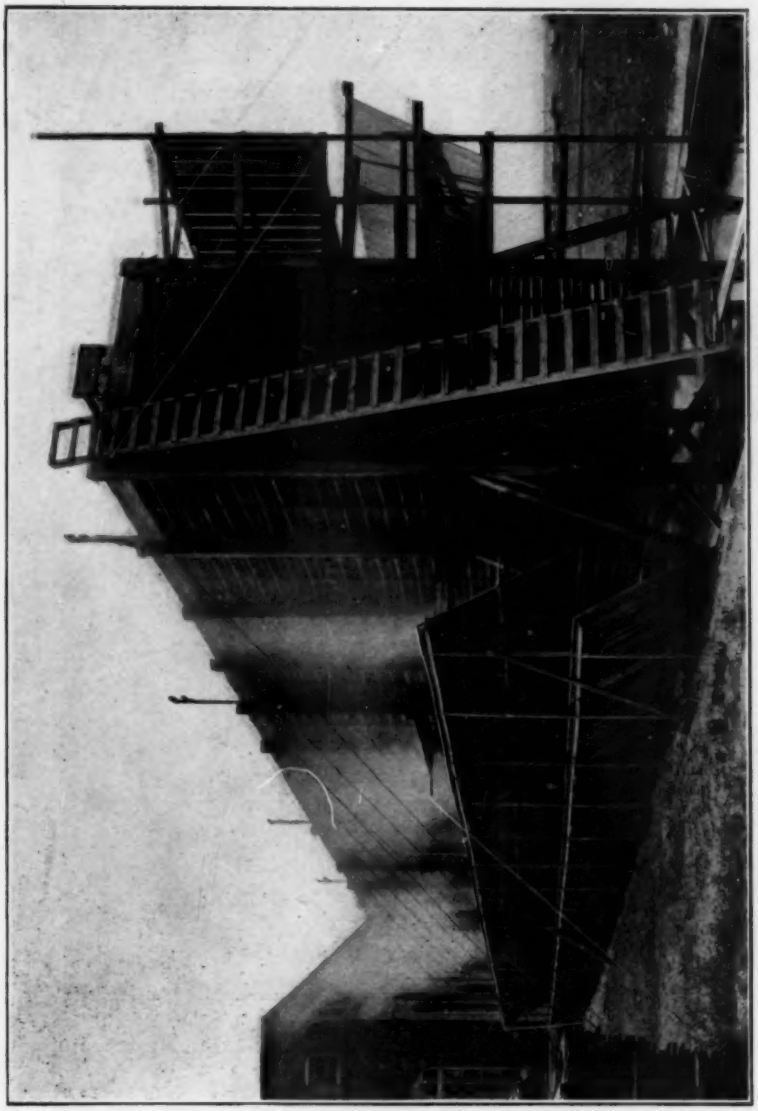

FIG. 4 2900-H.P. OPEN-TYPE WOODEN TOWER AT COLORADO SPRINGS FOR A LOW-PRESSURE TURBINE INSTALLATION WITH A LEBLANC CONDENSER

THE WIND SHIELDS REDUCE THE LOSS OF VAPOR DURING HIGH WINDS. COOLING, FROM 20 DEG. TO 30 DEG. HUMIDITY GENERALLY LOW—ABOUT 50 PER CENT.

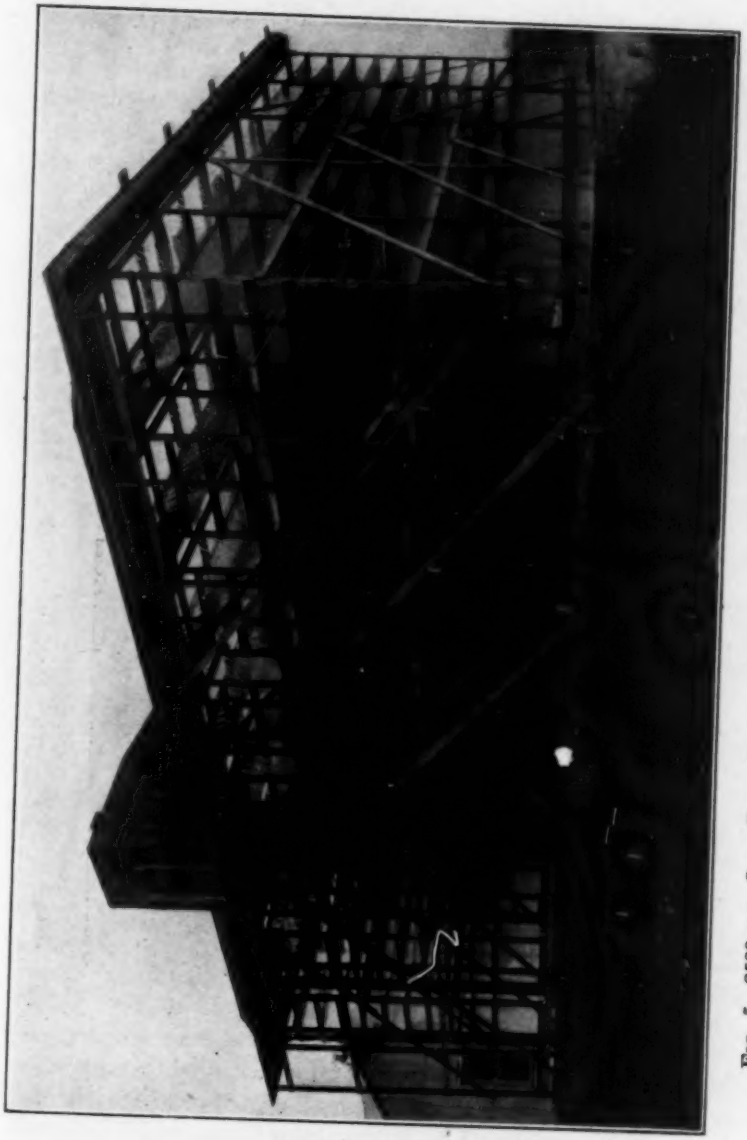

FIG. 5 2500-H.P. OPEN-TYPE SCREEN TOWER FOR A STEAM TURBINE PLANT AT VISALIA, CAL.

LOW—ABOUT 50 PER CENT.

ing. The expense for buying city water for this purpose would otherwise be prohibitive except in large plants.

REPRESENTATIVE INSTALLATIONS

6 As examples of present cooling-tower practice in connection with high-grade power properties, the following may be mentioned:

a At Gary, W. Va. (Fig. 2), three towers 25 ft. in diameter by 18 ft. high serve a plant of both high and low-pressure turbines

FIG. 6 5800 H.P. NATURAL DRAFT WOOD CONSTRUCTION TOWER FOR STEAM TURBINE PLANT AT BUTTE, MONT.

recently installed in connection with a non-condensing engine plant furnishing the mines with light and power. A unique feature is the induced-draft fan located in the stack and driven by a Pelton water motor, which is served in turn by a small turbine-driven centrifugal pump in the

power house. This equipment will be referred to later in connection with the combined natural-forced-draft type.

b The 13th & Mt. Vernon station of the Philadelphia Rapid Transit Company is another low-pressure turbine plant employing forced-draft cooling towers.

c At the central station at San Luis Potosi, in the Mexican highlands (Fig. 3), separate cooling towers serve engine

TABLE 2 WEATHER CONDITIONS, BUTTE, MONT.

AVERAGE TEMPERATURE AND HUMIDITY, 1894–1904 (WEATHER BUREAU, HELENA, MONT.)

MONTH	TEMPERATURE DEGREES	HUMIDITY PER CENT	MONTH	TEMPERATURE DEGREES	HUMIDITY PER CENT
January.....	23.7	66.8	July.....	62.8	45.3
February....	23.9	68.6	August....	62.9	43.4
March.....	28.3	60.8	September.	52.0	53.6
April.....	40.0	54.3	October...	45.1	54.4
May.....	49.0	54.9	November.	33.7	62.7
June.....	52.0	50.0	December .	26.5	70.5
Av. for year	41.6			58.9	

TEMPERATURE RANGES

BETWEEN DEGREES	DAYS	ABOVE DEGREES	DAYS	PER CENT YEAR	HOURS	PER CENT YEAR
70– 75	25	70	99	27	276	3.15
75– 80	24	75	74	20	196	2.22
80– 85	28	80	50	13.7	158	1.8
85– 90	19	85	22	0.6	51	.58
90– 95	2	90	3	0.82	9	.097
95–100	1	95	1	0.28	1	.01
		Total	249	67	681	7.9

Data from M. H. Gerry, Jr.

and producer systems, the make-up water being furnished from deep well pumps. This plant has been in operation since 1904.

d Western plants of inexpensive construction are the Mt. Whitney Power Company (Fig. 4), and the Colorado Springs Light & Power Co. (Fig. 5), both turbine plants.

e Perhaps the best example of the adaptability of cooling towers is an equipment designed and built by the Helena

Power Transmission Company for its auxiliary turbine station at Butte, Mont. (Fig. 6).

7 This latter tower was made the subject of an exhaustive preliminary study, and subsequent test, by the company's engineering organization, and through the courtesy of M. H. Gerry, Jr., chief engineer and general manager, the writer has been able to place the complete report at the disposal of the Society for future consideration. The tower (shown in Fig. 6) serves a turbine plant of 4000-kw. capacity at 28-in. vacuum. The designers state that after two years' experience the results coincide closely with the theoretical deductions made before its construction.

SPECIAL PHASES OF COOLING TOWER OPERATION

8 Two important factors fortunately contribute to the effective operation of a cooling tower:

a One factor is the well-known characteristic of a natural-draft tower considered as a "chimney"—increase in capacity with increase in temperature head (see Fig. 12).

9 In steam work, especially with high vacuum, the general range of condenser-discharge temperatures is relatively low; in gas work, on the other hand, it is high. Pistons are today operated at temperatures of 140 deg. to 160 deg., cylinders from 120 deg. to 150 deg., occasionally higher. Owing to the small volume of water in the minor circuits, such as valves, packings, etc., these temperatures have little effect upon the average outlet temperature of the engine, which ranges from 115 deg. to 130 deg. in the large engines, and 140 deg. in the smaller sizes and verticals. This would correspond to a very poor vacuum in a steam plant, not more than 24 in. to 26 in.; practically out of the question in turbine work. However, this high temperature results in a high rate of heat dissipation in the tower per unit of cooling surface, with a corresponding reduction in bulk of tower.

b The second factor relates to developments in the efficiency of the steam-condensing plant.

10 The function of a condenser is, primarily, that of a water heater and the measure of its efficiency as a condensing vessel is the difference between the temperature of the exhaust steam and that of the discharge water. A theoretically perfect condenser would heat the outgoing cooling water exactly to the temperature of the incoming steam. But in practice from 10 deg. to 50 deg. difference exists,

depending upon the type of condenser and the volumetric ratio of water to steam. A good surface condensing plant with dry-air pump should operate at 28-in. vacuum with a temperature difference of 15 deg.; often it is more, and the author has seen 25 deg. to 40 deg. difference in some of the largest stations in the country. A good barometric or centrifugal jet condenser, with dry-air pump, should operate with a temperature-difference of 10 deg. to 15 deg. Although it is possible for this type to operate on less—perhaps 5 deg. to 10 deg.—commercial practice rarely concedes such results.

11 A very recent development in air pumps has made it possible to operate on still smaller difference, from 2 deg. to 5 deg., with a

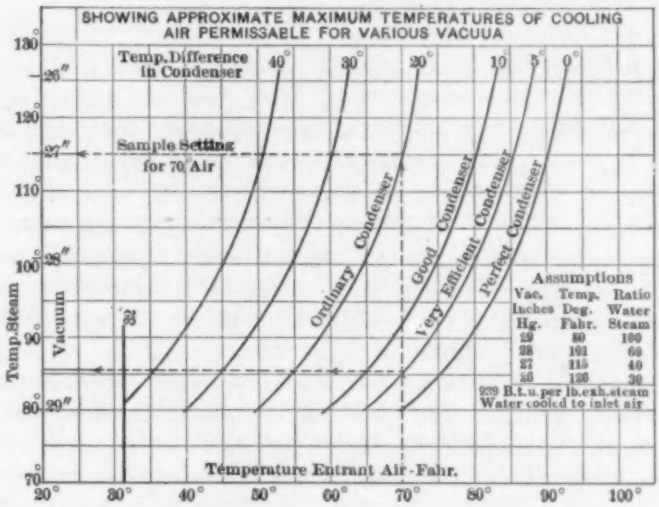

FIG. 7 SHOWING MAXIMUM AIR-INLET TEMPERATURES FOR VARIOUS VACUUMS

reasonable water ratio, and even to approximate theoretical conditions. All this is in the right direction. The smaller this temperature differential, the higher the maximum inlet temperatures permissible for a given set of conditions—both water to condenser and air to tower. The curves in Fig. 7 show this relation in approximate form—vacuum possible with varying condenser and fixed cooling tower performance. For example, with 28-in. vacuum, 20 deg. differential in the condenser, water ratio 60, and 15 deg. cooling in the tower, the highest possible temperatureof out side air would be 65 deg. Fahr. With warmer air, the vacuum would necessarily fall.

Under the same conditions, with 10 deg. differential, a maximum air temperature of 76 deg. would be permissible; and with 5 deg., 81 deg. inlet air. It is therefore apparent that the tendency of modern condenser development toward higher efficiencies will materially assist in the successful operation of cooling towers under extremely adverse conditions.

ELEMENTS OF DESIGN

12 The most important elements entering into the design may be considered under the following heads; having special reference to the enclosed type of cooling tower, which for a given floor space has by far the greatest cooling capacity:

a Type of cooling surface,

b Water distribution system.

c Draft and air distribution.

13 Following are a number of essential points that seem to the writer to have a most important bearing upon any type of tower designed for maximum duty and efficiency.

a All tortuous or unduly obstructed passages should be avoided. It is of no advantage to give ample spacing in one part of the tower and contract it in another, unless sufficient stack height is provided to overcome the additional resistance.

b Avoid free falling water. It should be distributed so as to descend clinging to some form of wetted surface.

c Avoid open spaces in the mat work, usually occurring at points where it is difficult to fill in between the frame of the tower. This will "short-circuit" and invariably diminish the effectiveness of the working sections.

d Reduce working section to minimum possible height, adding extra stack if necessary. The power required to elevate the water is important, and the working height of the tower is lost, even in a closed-condenser circulating system.

e Baffles or variable spacing are often necessary to obtain uniform air distribution.

f A settling basin of liberal depth is always advisable in order that entrained air may separate. In all jet-condenser installations, this is extremely important owing to the amount of air returned to the condenser; and even in

surface installations, this air will find its way back to the condenser via the feed water: result, impaired vacuum.

g All wooden mat surface is subject to swelling. Means should be taken to insure permanent alignment; otherwise serious reduction in draft area and capacity may be encountered.

h For maximum effectiveness, a cooling surface is required which provides uninterrupted descent of water, in a thin film at all times in intimate contact with ascending air. If any interruption is necessary, the descending sheet should be guided into place to avoid free fall.

PRESENT TYPES

14 The various types of cooling systems now in use naturally group themselves into a few general classes:

a The simple spiral-spray nozzle discharging into an open pond.

15 A prominent example is the 10,000-kw. Wyoming Ave. turbine station of the Philadelphia Rapid Transit Company, where this cooling pond is employed during a portion of the summer months. It has been suggested that the sprays be mounted upon the power-station roof, thereby taking advantage of the inclined surface of the roof for extra cooling effect, suitable gutters returning the water to the cold well. There might be some hesitancy about installing a reservoir on the roof; but in one notable instance, the recently designed gas-power station of the Duquesne Lighting Company, Pittsburg, Pa., the roof reservoir forms a very effective part of the cooling system. Here a small cascade type of tower assists in cooling. Without other agency this simple system requires only 10 to 20 per cent make-up water.

b The simple tray type, Fig. 3, with water dripping through perforations, and cooling entirely by means of transverse air currents from the side.

16 Here no direct draft is possible, and the tower has no direct cooling surface. The trays operate simply to arrest the fall of the water. In this respect, the type is a simple mechanical refinement of a rough frame tower filled with brush, such as has often been employed in temporary power work. It is, however, comparatively inexpensive, and under some conditions, may be utilized to advantage. The tower of the Potosina Electric Company, San Luis Potosi, Mexico, is built entirely of structural materials and cools from 10 deg.

to 30 deg. with very low humidities. Although encased in netting to prevent loss by spraying, as much as 10 per cent of the volume passing through the tower is carried away during a brisk wind. The inexpensive construction is shown in Fig. 5, using horizontal wire screens instead of perforated trays and without wind screens.

c The simple cascade type, constructed either of wood or of corrugated sheet, in which a considerable part of the cooling is by actual conduction.

17 In the case of the original gas engine service plant of the Union Switch & Signal Co., Swissdale, Pa., this cascade system materially assisted in the work of cooling the gas engine jacket water, but the abstraction of heat through the concrete walls of a large reservoir was largely responsible for the cooling. This cascade system seems to have been overrated. In one prominent plant, the author understands it to have been a decided failure; in any form, it is extremely primitive and not in accordance with effective design.

d Another representative of the simple types of construction is the multiple cascade. (See Fig. 8.)

18 Here the fall of water is simply interrupted at short intervals, and no cooling surface is installed. It is evident that successful operation is dependent entirely upon the accuracy with which the trajectory of the falling particles can be predetermined in the spacing of trays and maintained in the subsequent operation of the tower. This would require an absolutely constant head.

19 The tower at Colorado Springs (Fig. 4) utilizes the construction as in Fig. 8 *b*, a horizontal slotted surface with wind shields to prevent spray loss. This tower gives 40 deg. cooling in fair weather. The humidity however is very low, around 50 per cent (relative.)

e Several American towers are constructed simply of horizontal lattice work, usually of cypress, the numerous tiers being staggered in order to break more effectively the fall of water. (See Fig. 8 *c*.)

20 In some, the upper and lower faces of the lattice work are beveled to lessen the resistance of descending water and ascending air. Cooling water is distributed by atomizing nozzles, by numerous spray pipes, or by Barker's mill.[1] This type evidently does not lend itself readily to natural-draft work, owing to the serious resistance offered to the draft by the lattice work.

[1] Radial arm distributor propelled by lateral reaction of its own jets.

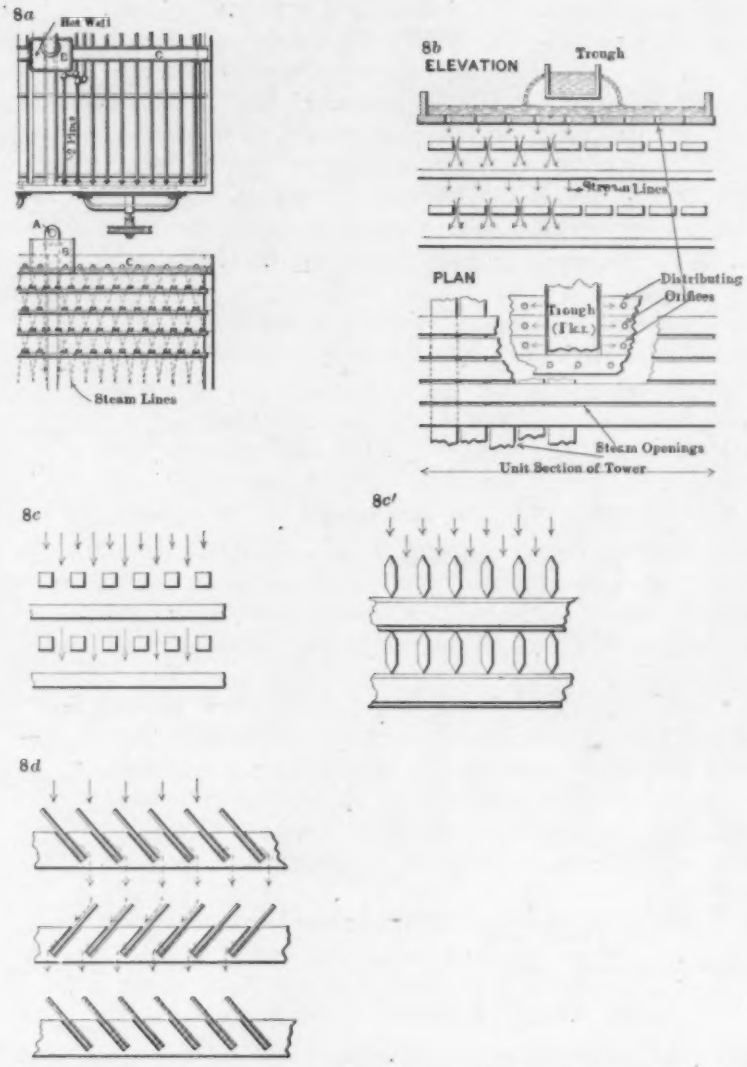

FIG. 8 *a, b, c, d* TYPES OF COOLING SURFACES EMPLOYED IN VARIOUS TOWERS

f A modification of the multiple cascade system, used in the German tower (Fig. 8 *d*), endeavors to utilize partly the inclined deflecting surface as a cooling medium, although it is a question whether this is of much effect owing to

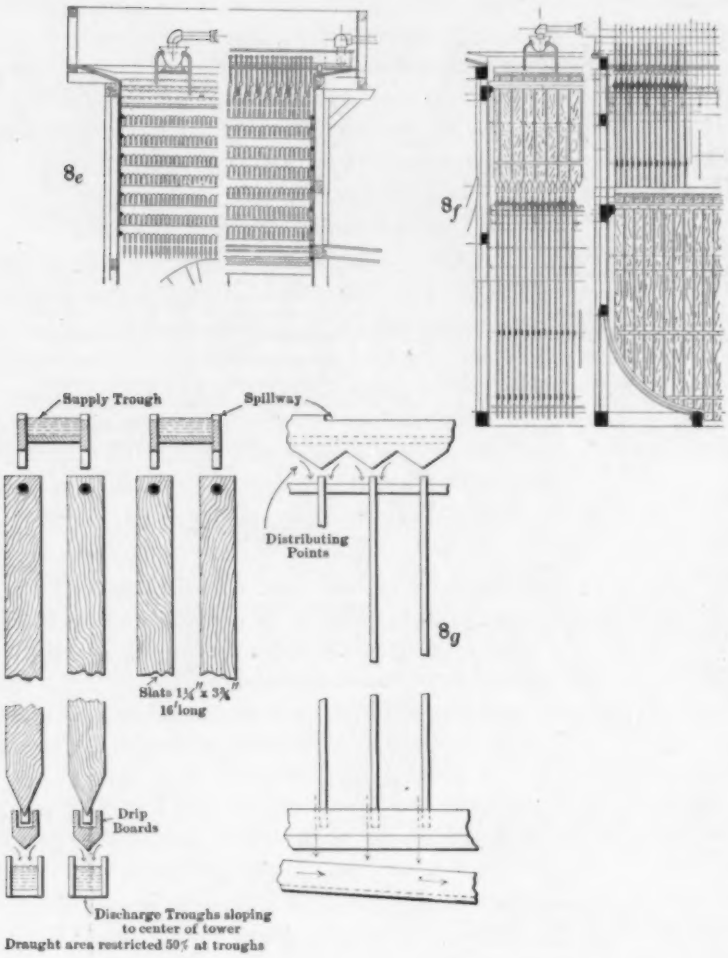

FIG. 8 *e*, *f*, *g* TYPES OF COOLING SURFACES

the fact that the ascending air in all cases impinges on the lower surface of the deflectors, and not on the upper wetted surface.

g Another German design (Fig. 8 *e*), which has been introduced into this country, advances one step in introducing vertical cooling surface in transverse tiers. But most important is the attempt to guide the water downward

in the form of a film, by forming each slat with a saw-tooth edge, meeting the lower transverse slats and guiding the water streams thereon.

21 The designer has evidently appreciated the necessity of avoiding free fall of water and deliberately piped the water from the flume to small troughs serving the upper row of slats.

22 An American builder has modified this system (Fig. 8 *k*) by practically discarding the numerous tiers of slats for vertical ones extending halfway down the tower, turning them 90 deg. at the middle of the tower, ostensibly for the purpose of equalizing air distribution. However effective this may be, we have here the desirable elements of continuously wetted surface and no free fall of water.

h A well-known American type, resembling the multiple-surface German tower (Fig. 8 *e*), employs numerous tiers of galvanized iron or tile cylinders, with a distributor at the top, of the Barker's Mill type, propelled simply by reaction of the issuing jets.

23 Although highly effective in fan-type towers, there is much free falling water owing to the non-continuity of cooling surface; and in the author's experience, there is some objection to the Barker's Mill distributor in the difficulty of maintaining ball bearings in proper condition.[1]

i Several builders employ continuous galvanized-iron surface from top to bottom of tower, either in the form of corrugated sheathing or of wire mesh, the water being carefully guided to the sheets so as to avoid free fall. The principle is right. With the close spacing permissible, a most intimate contact of air and descending film may be maintained.

j Coming now to exclusively wooden-mat construction, an example of the attempt to combine in a single slat construction all the above-mentioned desirable features, is that shown in Fig. 8 *g*.

24 This resembles the form employed by Mr. Gerry, at Butte, Mont., although it is sketched from a design by Mr. Moser, of the Newhouse Mines & Smelters Co., Newhouse, Utah. This tower is

[1] In a Detroit station, the entire condensing plant lost its vacuum on several occasions at peak load owing to the stoppage of this distributor; and, finally, a three-deck phosphor-bronze ball bearing had to be designed to withstand the corrosion from the ascending vapor.

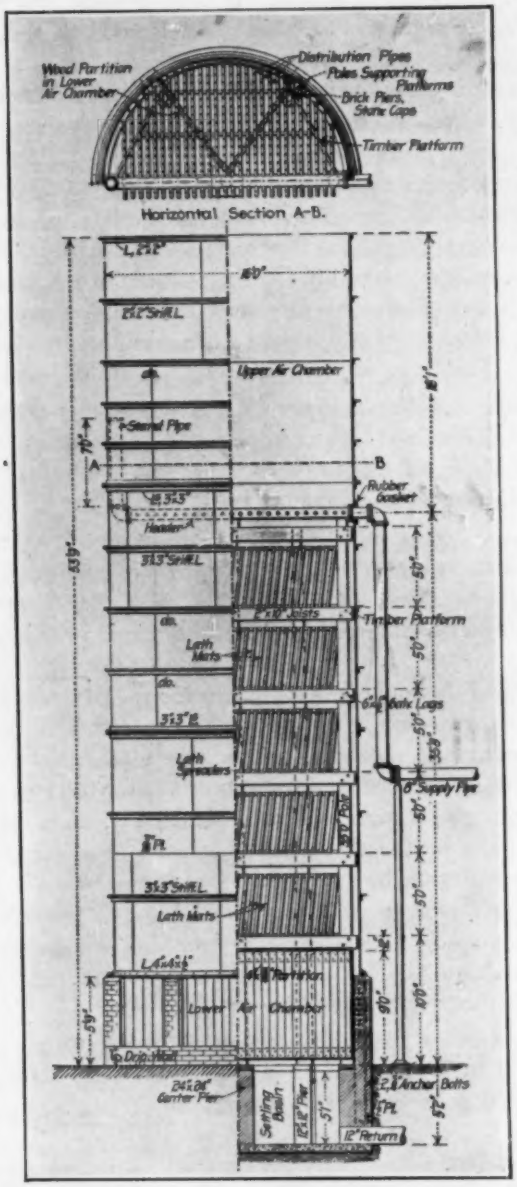

FIG. 9 EXPERIMENTAL NATURAL-DRAFT TOWER AT DETROIT, MICH.

of the natural-draft type with a side flume communicating with numerous transverse ducts which discharge upon continuous vertical slats, the saw-tooth construction being employed to guide the water on to the wetted surface. At the bottom, instead of allowing the water to fall freely into the receiving basin, each descending sheet is caught in a small trough and conveyed to the center of the tower, where it descends without retarding the ascending current of air. These distributing troughs reduce the effective draft area of the tower by about 40 per cent; but, on the other hand, the reduction in area is fairly uniform throughout the tower, and the area correspondingly diminished. That this type is extremely effective is proved by the results of the tests at Butte.

LATH MAT CONSTRUCTION

25 In a design originated in Detroit, Mich., shown in detail in Fig. 9,[1] an attempt was made to subdivide the cooling surface into sections or tiers, while maintaining the advantages of continuous vertical surface. This it was thought would facilitate the construction and repair of the tower; it was also hoped to avoid the distortion of the mat surface occasioned by the swelling[2] of the timber, which it is hard to avoid when long slats are employed.

26 This tower was designed under the direction of Alex Dow, Mem.Am.Soc.M.E., general manager of the Detroit Edison Co., largely in order to try out the natural-draft type under conditions of central station operation. There were many features which could have been improved upon, but that the type of mat surface employed was extremely effective is shown by the results of tests made in 1902.[3] The important point in design was reduced cost of construction. With the exception of the sheet-steel shell furnished by a local boiler

[1] Described in *Engineering News*, March 20, 1902.

[2] For example, this difficulty was experienced in a large gas-engine cooling tower in Texas. Plain horizontal platforms were used, with boards spaced far enough apart for the water to drip through. After some time in service, the timber had swelled to such an extent as practically to close off two-thirds of the tower, deflecting the greater portion of the water to the sides, where it descended without being cooled to any extent. This trouble, of course, is not so serious in the vertical slat tower; yet in a Boston plant employing rough boards set vertically on edge, the boards so swelled and warped that they practically closed the intervening air passages at certain points. This distortion may be noted somewhat in the Colorado Springs tower (Fig. 5).

[3] Conducted by the author and by Messrs. Armstrong and Richardson.

maker, the tower was built by unskilled labor employed about the station, and its total cost, including shell, concrete and brick work, material and labor, was in the neighborhood of $1350, serving a 1000-h.p. engine-driven plant. The shell was designed self-supporting with an independent internal frame work for supporting the weight of the mass. Wooden sheathing could have been used to good advantage, however, and the entire tower constructed by unskilled labor.

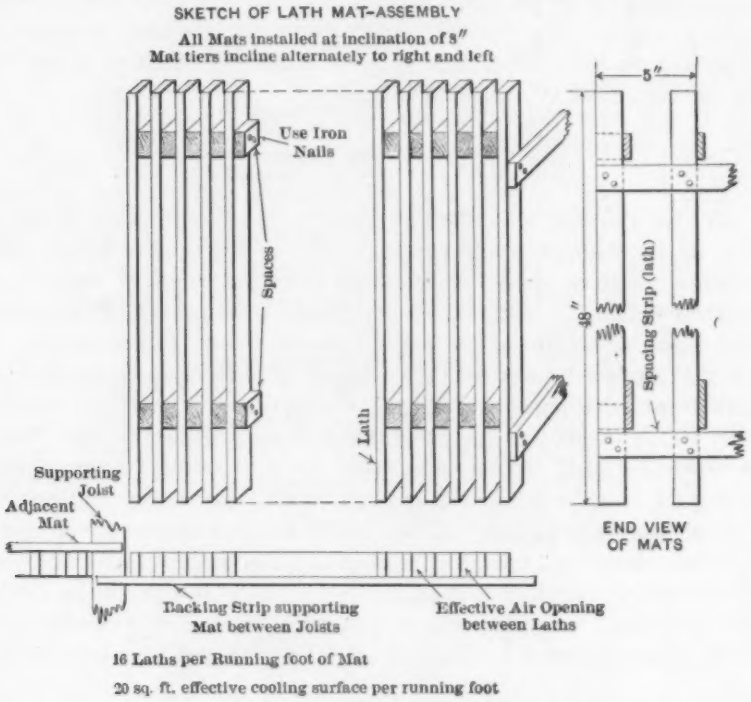
SKETCH OF LATH MAT-ASSEMBLY
All Mats installed at inclination of 8″
Mat tiers incline alternately to right and left
Use Iron Nails
Spaces
Lath
Supporting Joist
Adjacent Mat
Backing Strip supporting Mat between Joists
Effective Air Opening between Laths
5″
Spacing Strip (lath)
8″
END VIEW OF MATS
16 Laths per Running foot of Mat
20 sq. ft. effective cooling surface per running foot

FIG. 10 *a* DETAILS OF SECTIONAL MAT SURFACE

27 The mat surface was constructed of common wood lath, assembled on a form, with iron nails protected from corrosion by being imbedded in the wood. Mat details are shown in Fig. 10 *a*. These lath mats produced a very desirable form of cooling surface. The rough surface kept the descending stream in constant agitation, and there was sufficient slope to prevent free falling water for any great distance, and also to constrain the ascending air to slice upward through the interstices, thereby bringing into use both sides as well

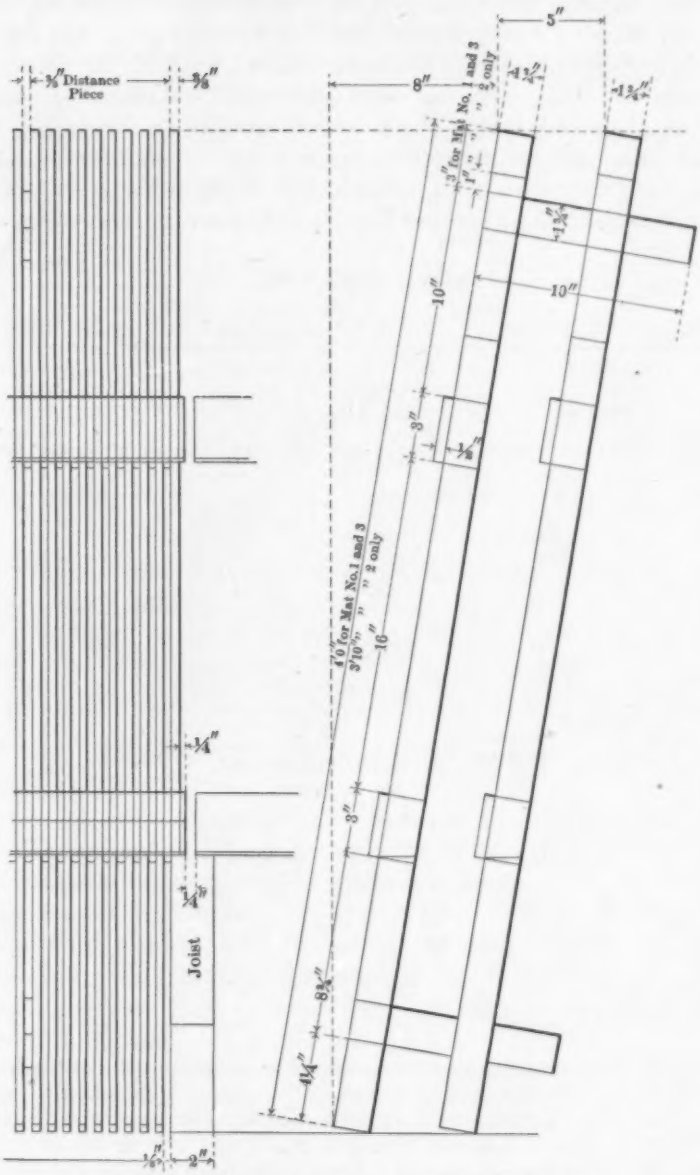

FIG. 10 *b* DETAILS OF SECTIONAL MAT SURFACE

as both edges of the lathe. Thus a cooling surface of approximately 20 sq. ft. per running foot of lath mat was obtained. The various tiers were readily assembled in succession, working from the shell inward until full. Uniform water distribution was effected by means of the pipe-spray system, with laterals spaced like the mats below.

28 Two series of tests[1] were made at Detroit at different times, first, with only the two upper tiers, and finally with all the mats in position. Tables 4 and 5 and Figs. 11 to 13 show the relation between

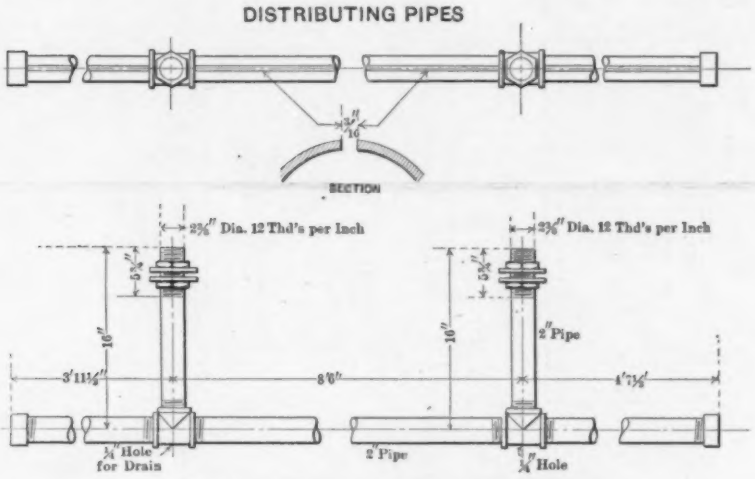

FIG. 10 *c* DETAILS OF DISTRIBUTING PIPES

the various quantities observed. It was very noticeable that the rate of heat dissipation in B.t.u. per sq. ft. per hr. was considerably higher for the uncompleted tower with only about three-fifths of the mats in operation. However, by the addition of the remaining mat surfaces the tower was enabled to work on lower water temperature, and we should therefore expect a lower rate of heat dissipation. This would indicate that the upper tiers of towers were more effective

[1] In this plant the condensing system was not well adapted to economic working. Air and circulating pumps were direct-coupled, making it impossible to control the tower water separately from the condensation. There was considerable air in the system from a long run of exhaust piping; and with no dry-air pumps, a vacuum of 24 in. was normal practice. But the condenser was operated with a temperature differential of 47 deg., so that *with an efficient condenser*, a vacuum of 28 in. might have been obtained with the same tower performance, 16 deg. cooling, 71 deg. cold well.

TABLE 3 COMPARATIVE DATA, DETROIT EDISON TOWERS

TYPE	FORCED DRAFT STATION *A*	NATURAL DRAFT STATION *C*
Rated Engine, i.h.p.	1500	900
Cooling surface, sq. ft.	34,780	24,500
Surface per h.p.	23.2	27.2
Space occupied, cu. ft.	7064	10,850
Space occupied, sq. ft.	175	200
Cost complete	$3900	$1350
Cost per h.p.	$2.60	$1.50
Auxiliaries, e.h.p.	13	..
DIMENSIONS		
Delivery pipe above ground	29 ft. 0 in.	35 ft. 8 in.
Height over all	40 ft. 6 in.	53 ft. 9 in.
Height mat section	17 ft. 0 in.	25 ft. 0 in.
Height stack	12 ft. 6 in.	18 ft. 1 in.
Height outlet	10 ft. 10 in.	9 ft. 0 in.
Diameter tower	14 ft. 10 in.	16 ft. 0 in.
Diameter fan	9 ft. 3 in.	..

TABLE 4 TEST OF NATURAL-DRAFT COOLING TOWER, DETROIT

INCOMPLETE, THREE-FIFTHS SURFACE INSTALLED

TIME	TEMPERATURE, DEG. FAHR.					QUANTITIES				
	AIR	HOT WELL*	COLD WELL	WATER COOLING	TOTAL HEAT HEAD†	TOWER WATER LBS. PER HR.	HEAT DISSIPATED B.T.U. LBS. PER HR.	HEAT PER SQ. FT. COOLING SURFACE B.T.U. PER HR.	CIRCULATING WATER PER SQ. FT. LBS. PER HR.	LOAD KW.
1	2	3	4	5	6	7	8	9	10	11
12 noon	34	102	89	13	68	375, 000	4,880,000	332	25	270
1.30	35	106.5	90	16.5	71.5	‡ 375,000 370,200	6,108,000	415	24.8	315 290
2.30	35	106.5	87.5	19	71.5	375,000	7,120,000	484	25	315
3.30	35	113	88.5	24.5	78	375,000	9,000,000	613	25	350
4.30	32.5	100	84	16	67.5	399,000	6,384,000	434	26.6	365
5.00	28.5	103.5	88	15.5	75	445,500	6,900,000	470	29.7	485
6.00	26	125	94	31	99	417,000	12,930,000	880	27.8	655
7.00	24	121	94	27	97	427,000	11,532,000	785	27.4	570
8.00	24	123	94.5	28.5	99	427,000	12,174,000	827	27.4	600

*Assuming a more efficient condenser, say 10 deg. difference, the probable vacuum would be 26 deg. to 27.5 deg. This condenser actually operated at 40 deg. to 50 deg. difference.

†Total heat head = air heating + lost head.

‡Only three-fifths cooling surface installed.

§Difference due to rapid change in load.

TABLE 5 RESULTS OF TEST OF NATURAL-DRAFT TOWER, DETROIT

COMPLETE, FIVE-FIFTHS SURFACE IESTALLED

Engines:	Two 400 i.h.p. 300 kw. MacIntosh & Seymour tandem compound engines, overhung generators.				
Condensers:	Worthington surface (admiralty type) 1600-sq. ft. reciprocating wet-air pump and circulating pump.				
Tower:	Wood mat construction, 24,500 sq. ft. evaporating surface, exclusive of shell.				
Test:	March 15 to 16, 1901, 4 p.m. to 4 p.m., 24 hr.				
		A.M.	P.M.		AVERAGE
Weather:	Barometer (abs.), min.	30.22	30.07;	30.14	30.27
	Temperature air, deg.	18.5	25;	30	25
	Relative humidity, per cent	76	82;	58	72
Load:	600 kw. max. to 50 kw. min. Average				244.9 kw.
	Engine Efficiency = 92.5 =875 i.h.p. max. Average..				354.8 i.h.p.
Steam:	Weight of condensed steam per hr.,lbs.....				5910.6
	Temperature exhaust steam, deg. Fahr.....				134.38
	Temperature condensed steam, deg. Fahr.....				108.78
	Weight of steam per hr., max. load, lbs.....				13,500
	Vacuum (abs.) 25 to 19, average about.....				22
	Vacuum corresponding to temperature exhaust steam....				25
	Vacuum possible with good condenser (10 deg. difference).				28
Water:	Circulated per hr., lbs.....				293,536
	Temperature hot well, average, deg. Fahr.....				87.50
	Temperature cold well, average, deg. Fahr.....				71.27
	Vaporization loss per hr., lbs.....				5,970
Results:	Condenser surface per kw., sq. ft.....				2.66
	Steam per kw. hr., lbs.....				24.3
	Steam per i.h.p. hr., lbs.....				16.66
	Circulating water per lb. of steam, lbs.....				49.6
	Steam per sq. ft. condenser surface per hr., lbs.....				3.7
	Circulating water per sq. ft. tower surface, lbs.....				12.0
	Difference in temperature between exhaust steam and discharge, deg. Fahr.....				47
Cooling:	Max. 20 deg., min. 3 deg–5 deg. Average				16.23
	Heat dissipated per hr, B.t.u.....				4,769,000
	Heat per sq. ft. tower surface, B.t.u.....				195
	Heat per sq. ft. per 1000 lb. water, B.t.u.....				0.665
Evaporation:	Circulating water, per cent.....				2.03
	Engine steam, per cent.....				101
Tower:	Surface per kw. (average load 245 kw.), sq. ft.....				100
	Surface per kw. (max. load 600 kw.), sq. ft.....				408
	Surface per 1000 lb. steam max. load, sq. ft.....				1.82
	Surface per 1000 lb. steam average load, sq. ft.....				4.14
	Surface per 1000 lb. circulating water per deg. max. cooling, sq. ft.....				5.22

than the lower. The heat dissipation during the tests on the complete tower ranged from 200 B.t.u. to 300 B.t.u. per sq. ft. of surface per hour under normal conditions, and this could undoubtedly have been increased in a carefully constructed tower with suitable condenser apparatus.

29 In general, the tower showed very little difference in efficiency summer and winter, rather against expectations. Apparently the increased evaporation possible in the higher air temperatures of summer offset the greater conduction of heat in the colder air of winter. In very hot weather, a negligible effect from conduction was apparent, from the fact that at certain maximum loads the vacuum fell rapidly, indicating that the capacity of the tower had been reached, due to complete saturation of air, while in cold weather the vacuum would hold up better at the same load. This shows that with air fully saturated and evaporation checked, the dissipation of heat by conduction in hot weather was quite insufficient to give an appreciable margin of overload.

30 This tower was in constant use for a period of about four and a half years, cooling all of the condensing water for the central station. Depreciation was at first thought to be a serious factor, but later, when the tower was finally dismantled, the frame work and mats were found to be in excellent condition. The only parts showing deterioration were the upper sheets of the tower shell lying above the distributor, where corrosion had taken place owing to the alternate wetting and drying of the surface during the last six months of service in 1905 (10-hr. operation). The mats themselves were as sound as when put in. After a few months' service, the mat surface became coated with scale due to the incrustating properties of the water. This scale would accumulate, crack off and fall to the settling basin.

31 Although fairly successful, this experimental design might have been considerably improved. By straddling the supporting joists in the manner shown in detail in Fig. 10 *b*, the various tiers of mats may be brought together into a practically continuous surface from top to bottom, thus entirely preventing the fall of water (Fig. 12). At the bottom, the obstruction to draft may be prevented by employing deflecting troughs under each mat, to convey the water to the center of the tower, as in the Moser tower, Fig. 8 *g*. A better distribution system in the form of horizontal slotted laterals discharging upward and over-flowing directly on to the respective mat sections is shown in Fig. 10 *c*.

32 In any system of stationary jets, it is extremely difficult to

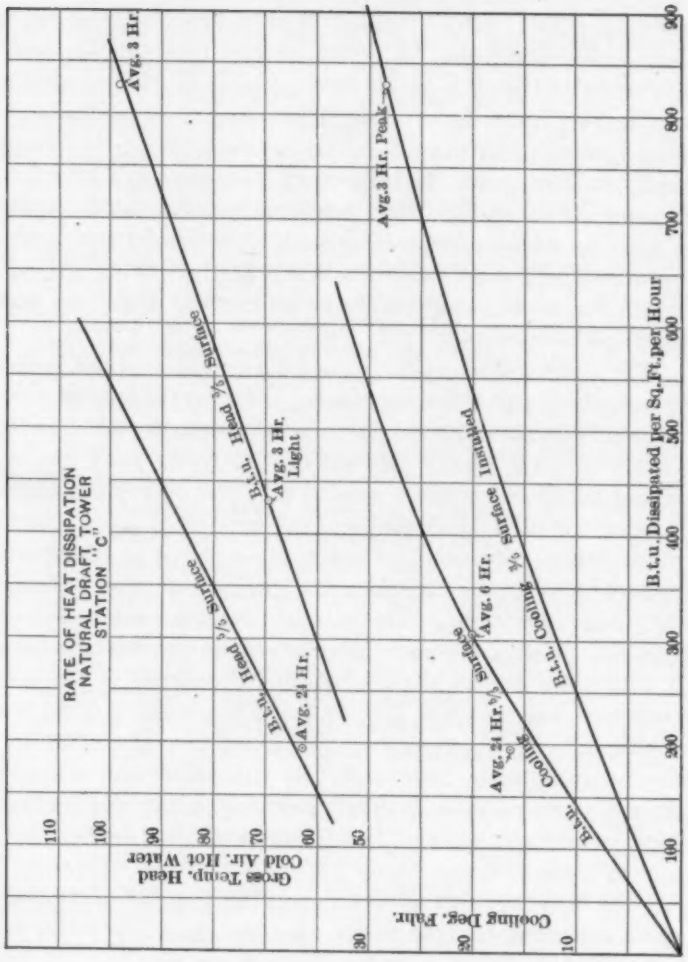

FIG. 11 RATE OF HEAT DISSIPATION OF THE DETROIT TOWER

obtain uniform distribution of water over the entire tower at different rates of flow. With the slotted pipes, it is an easy matter to open or close the slots so as to distribute uniformly, and as they are laid horizontally, this adjustment is permanent. It is also easy to free the laterals from foreign matter, as is not the case with jets. This is because of the ample section of the laterals; whereas any system using a large number of small distributing pipes or apertures involves ultimate trouble from clogging.

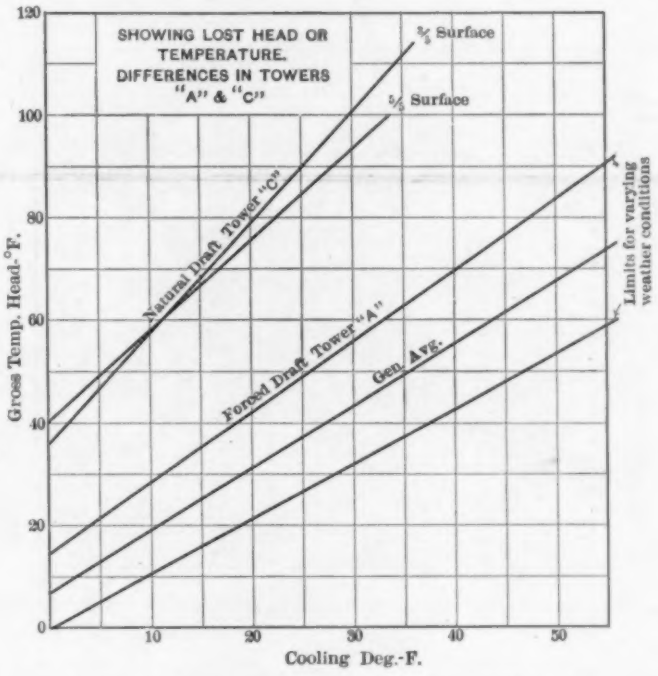

FIG. 12 LOST HEAD OF NATURAL AND FORCED-DRAFT TOWERS

THE EVAPORATIVE COOLER

33 In gas-engine work it is often necessary to economize water to the greatest possible extent. In an Arizona mining plant employing gas engines, where the mine water was so foul and acid as to prohibit entirely its use for cooling jackets, an evaporative cooler was recently constructed of ordinary hot-water radiators arranged in series-parallel, with air forced over the surface by a motor-driven

fan. The well-known counter-current system was employed, and the outfit was fairly efficient, the jacket water being cooled 15 deg. with a power consumption of 5 per cent of the output of the engine.

34 It occurs to the author that by keeping the radiator surface continually wet the effect of evaporation as well as convection might be utilized in cooling. The foul mine water may sometimes be used for this purpose without contaminating the jacket circulation. With an expenditure of $2\frac{3}{4}$ per cent in evaporation, an increase of 24 per cent in cooling would be obtained, assuming the air entering and

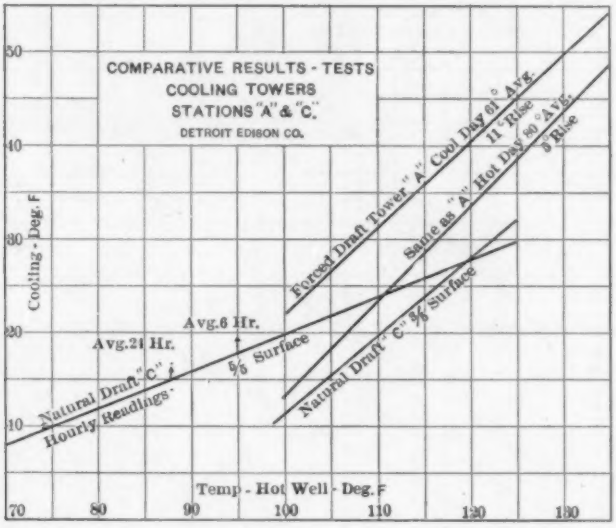

FIG. 13 COMPARATIVE RESULTS, NATURAL AND FORCED-DRAFT TOWERS

leaving to be fully saturated (Fig. 15). This system has been attempted in connection with steam condensers, but apparently without much success. The principle seems entirely logical, but the difficulty of maintaining tight joints with thin-walled tubes of sufficient diameter to permit of the passage of the proper amount of air, would seriously detract from the effectiveness of this apparatus by reason of air leakage. The low vacuum shown during tests of such apparatus largely confirms this supposition. For gas-power plants, however, the type seems admirably suited.

STANDARDS OF DESIGN

35 The cooling tower should be designed with the same flexibility as other good power-plant apparatus, as regards capacity under various conditions of operation; it is subject to the same peak-loads as the prime mover. As a matter of fact, relatively more heat must be

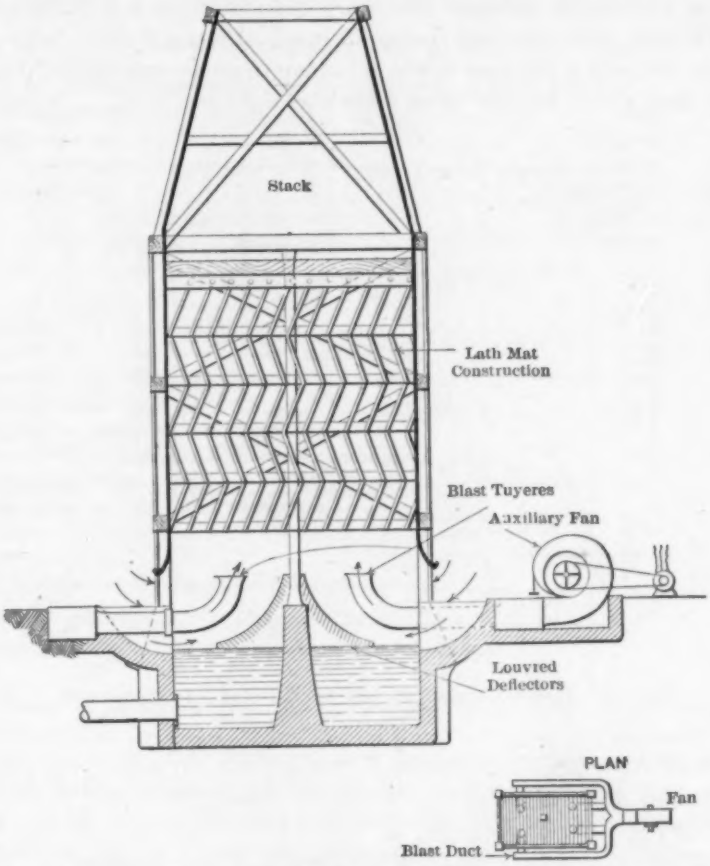

FIG. 14 PROPOSED COMBINATION TOWER WITH NATURAL AND AUXILIARY-FAN DRAFT

abstracted by the tower during peaks owing to the higher steam consumption of a steam engine per horsepower-hour on overloads. Consider, for example, a normal central station load. The evening peak seldom extends over three hours, and usually the most severe demands

on the generating apparatus occur within a period of one hour. Here, then, a definite overload capacity in the cooling tower is as desirable as in the engine or boiler; and some means should be employed to relieve during these peaks the tower which would have ample capacity to operate unaided during the remainder of the day.

36 Again, consider the comparatively short periods of unfavorable weather in normal climates. Reports[1] from the Butte plant (Table 2) reveal a mean temperature of 41.6 deg. fahr. Yet there were 99 days of the year in which the temperature was above 70 deg.; 50 days above 80, and three days above 90 deg. Fahr. Taking 70

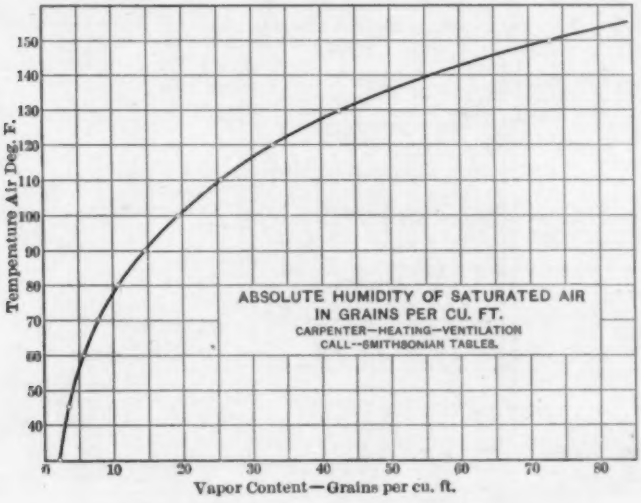

FIG. 15 VAPOR-CONTENT IN AIR AT VARIOUS TEMPERATURES

deg. as an empirical standard, it is apparent that for two-thirds of the year this temperature would not be exceeded. But careful *hourly* observations at Butte show that a temperature of 70 deg. was exceeded only for 681 hr. throughout the year; i. e., *7.9 per cent of the actual time.*

37 A study of Pittsburg conditions shows similar results (Table 1). Average throughout the year, 52.3 deg. The temperature during June, July and August averaged 10.5 days per month above 85 deg., and the humidity, 15.3 days per month above 80 per cent.

[1] Provided by Mr. Gerry.

Although the actual *hours* of maximum are not available from the Weather Bureau reports, it is safe to say that these unfavorable atmospheric conditions existed not more than one-tenth of the daily period, or 2.5 hr.

TABLE 6 OPERATING DATA, OPEN-SCREEN TOWERS

MT. WHITNEY POWER CO., VISALIA, CAL.

Tower designed for 1500 kw.—2000 kw. at 27-in. vacuum	
Horizontal screen surface, sq. ft.	9550
Circulating water handled, gal. per hr.	1,720,000
Rate of circulation, lb. per sq. ft. per hr.	1500
Dimensions, ft.	30 by 47 by 15 high
10 tiers galvanized iron screens	5 mesh per in.
Cost of tower including concrete form	$2,000
OBSERVATIONS, OCTOBER 23, 1906	
Maximum load carried, 5.20 p.m., kw.	1130
Temperature atmosphere, deg. Fahr.	55
Depression, wet bulb thermometer, deg.	8.5
Relative humidity, 50 per cent absolute, gr. per cu. ft.	2.35
Temperature incoming hot water, deg. Fahr.	110
Temperature outgoing cold water, deg. Fahr.	100
Cooling, deg. Fahr. (minimum for day)	10
Vacuum carried (ref. 30-in. barometer), in. Hg.	26.6
Difference between temperature steam and condenser discharge	
Possible vacuum (10 deg. difference in condenser)	
Maximum cooling for day (730 kw.), deg. Fahr.	16

Data from Hunt, Mirk & Co., Engineers, San Francisco, Cal.

38 It is apparent from the above that the problem of maximum capacity in cooling towers involves a condition of peak load existing only 5 per cent of the time, and high temperature only 8 per cent of the time. Moreover, these maximum demands will not generally occur at the *same hours* of the day. In the example cited in Par. 11, 5 deg. difference in the condenser, the maximum permissible air temperature would be 81 deg.; the more efficient the condenser, the higher the allowable air temperature. Yet at Butte, the temperature of 85 deg. was exceeded during only 22 days of the year, or 51 hr. This is equivalent to 2.5 hr. during mid-day and less than 0.6 of 1 per cent of the total time.

39 Is it, therefore, good engineering to design a cooling tower installation with a vacuum-producing capacity *large enough for any*

and all emergencies; or, on the other hand, to provide auxiliary means for assisting during these brief periods of maximum demand, while keeping the proportions of the tower within reasonable limits for normal operation? Might not even a considerable impairment of vacuum under the most unfavorable operation be better tolerated than the increased expense of equipment suited to maximum demand?

40 This, of course, applies particularly to natural-draft towers. Flexibility already exists in the forced-draft tower through the speeding of the fans; but even here there are some drawbacks owing to the high velocities already employed for normal working. Any large increase in the velocity of the fan may seriously disturb the uniformity of air distribution over the tower surface and give rise to eddies destructive to efficiency. That this condition exists, is very plainly shown by a survey of the discharge velocity by means of an anemometer. Examination of one defective installation by this method revealed the fact that fully one-third of the area was practically ineffective and that reverse currents actually took place in some parts. The air-distribution problem is exceedingly important, and more so in the forced-draft than in the natural-draft tower, where low velocities favor uniformity.

"BOOSTER" TYPE OF TOWER

41 The natural-draft tower is of itself ill-adapted for operating with a fixed temperature head. It thrives on the weakness of the condensing system. The lower the vacuum, the better the tower works, because of the increase in temperature head. And as this is clearly a problem of chimney design, the only way out of the difficulty is apparently by some method of auxiliary draft. As the specific heat of air is about 0.23, it is evident that an increase of 25 per cent in heat dissipation would require roughly double this increase in quantity of air, in order to maintain the same temperature conditions. This, however, is well within the capacity of a comparatively small fan auxiliary.

42 There are two methods of accomplishing this result:

a By locating in the stack an induced-draft fan whieh normally remains idle.

b By installing at the base of the tower a forced-draft system so designed as to supplement the natural draft without causing a back-flow.

43 An example of *a* is used in the cooling towers at Gary, West Va. (Fig. 2). Although designed for constant service, the arrangement of the stack fan is precisely as suggested. Whether the Pelton type of motor, direct-connected to the fan, is superior to belt or chain drive from a motor mounted on the outside of the tower, is a question of mechanical convenience: the essential elements are present.

44 The second suggested arrangement is crudely shown in Fig. 14. The auxiliary air is delivered to the tower through four "L"-nozzles supplied from a concrete duct surrounding the base of the tower. With this arrangement, the natural draft under the base of

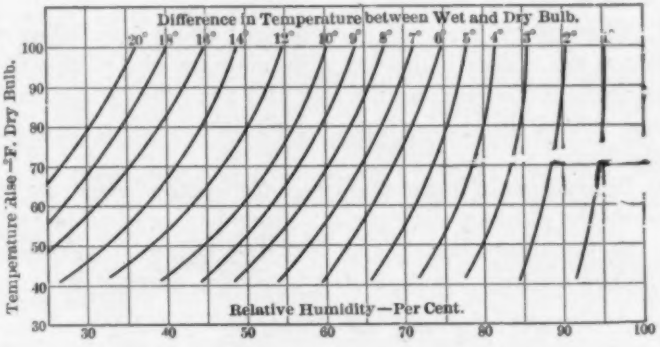

FIG. 16 HUMIDITY CHART FOR WET-BULB THERMOMETER

the tower might tend to be reversed owing to the back-pressure resulting from the blast. It is believed, however, that with a fairly open mat structure such as has been described above, the introduction of four auxiliary blast ducts would serve only to entrain more air and further assist the tower in the absorption of heat.

45 Of the two systems, the former has the advantage of being already put into practice. However, there is to be said in favor of the latter that no working parts, such as fan bearings, belt transmission, etc., are in the current of vapor; and with a tower operated intermittently, corrosion is an important matter, as was proved by the deterioration of the upper-sheets at Detroit. Furthermore this system lends itself more readily to a square or rectangular-shaped tower, which may be desirable in large sizes.

CONCLUSIONS

46 Improved types of cooling towers are in active demand for high-grade power plants both steam and gas, especially for low-pressure turbine installations.

47 Effectiveness of working is largely dependent on efficiency of condenser, i. e., minimum temperature difference between steam and discharge water is desirable to increase the temperature head on the tower.

48 Cooling towers are particularly adaptable to gas power plants. The bulk of the tower is reduced by the high temperature head available with hot jacket water.

49 Elements of most effective design; avoid free falling water; maximum retardation of descent with minimum obstruction of draft; insure uniform distribution of water and air; provide the maximum exposed wetted surface for a given bulk and an interrupted descent of fluid film.

50 In locations of low humidity simple forms of construction usually serve the purpose, except where ground space is valuable.

51 Sectional lath mat type of tower well adapted tonatural draft work. The construction suggested is simple, durable and inexpensive.

52 Normal rate of heat dissipation obtained by lath mat construction, 200 B.t.u. per sq. ft.

53 Auxiliary fan "booster" suggested as the best means of obtaining the desired overload capacity—a combination of natural and forced draft. Overload conditions (high temperature, humidity or load) usually last but a small percentage of the time—1 to 5 per cent. Natural draft suffices for the major portion.

THE PITOT TUBE AS A STEAM METER

By Prof. Geo. F. Gebhardt, Chicago, Ill.

Member of the Society

Steam meters may be conveniently grouped in two general classes, which, for lack of more suitable names, may be designated as *a*, series meters and *b*, shunt meters.

2 The series meter is an integral part of the piping, the entire mass of fluid to be measured passing through the apparatus. The St. John's and Venturi meters are the best known of this class. In the former the *volume* of fluid passing is determined by the rise and fall of a weighted plug valve and in the latter the *velocity* of flow is determined by the well-known principles of the Venturi tube. Both are indicating instruments and show only the rate of flow.

3 In the shunt meter only a portion of the steam to be measured is diverted through the apparatus, the velocity of flow through the shunt being an indication of that in the main pipe. In this class one or more small openings ½ in. or less in diameter suffice for attaching the apparatus to the pipe. One instrument suitably calibrated may answer for any size of pipe. The Pitot tube forms the basic principle of practically all meters of this class.

4 It is the object of this paper to describe a number of applications of the Pitot tube for steam measurements as constructed and tested at the Armour Institute of Technology.

5 The Pitot tube was first used by its inventor, Pitot, in 1837, in the measurement of the flow of water and since then has been successfully used in measuring the flow of many fluids and all true gases. Considerable difficulty has been experienced, however, in its application to vapors condensable under normal conditions of operation; and so far as the writer knows, no commercially successful instrument is on the market.

6 Many of the instruments are interesting laboratory devices and are of considerable value for experimental investigations; but on

All Papers are subject to revision.

account of the great number of variables involved, fall short of being practical commercial instruments. Fig. 1 illustrates the most common and the least accurate application of the Pitot tube for measuring the flow of condensable vapor in a pipe. S is the static nozzle at right angles to, and D the dynamic nozzle facing the current. U is an ordinary manometer partially filled with mercury. When there is no flow the mercury in columns N and W will be on the same level and the upper portions will be filled with condensed vapor. When there is a flow the mercury will be depressed as indicated and the difference in height H of the mercury columns in the two tubes will be

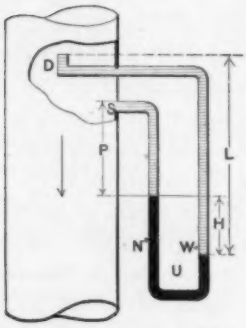

FIG. 1 PITOT TUBE WITH MERCURY MANOMETER

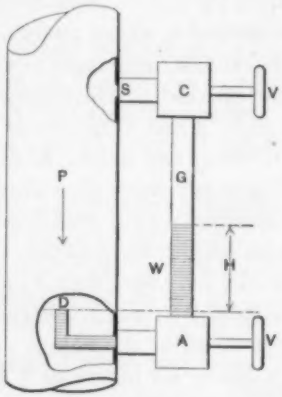

FIG. 2 SIMPLE GAGE-GLASS METER WITH SELF-ADJUSTING WATER COLUMN

a measure of the velocity of flow in the main pipe. On account of the great density of mercury and the variation in height of the condensed vapor above the mercury this application of the Pitot tube has very little value scientifically or commercially. For example: With dry steam at 100 lb. gage, a velocity of 8000 ft. per min. would give a depression H of only one in. and an error of 1/100 in. in measuring H would mean an error of 40 ft. per min. in the velocity.

7 In Fig. 2 is shown the original apparatus designed by Prof. R. Burnham of the Experimental Department of Armour Institute of Technology and the writer, in which the water of condensation is used as a self-adjusting column in place of mercury. This embodies the basic principle of many of the meters constructed later.

8 Referring to Fig. 2, A and C are two ordinary water gage cocks

and G an ordinary glass tube. Gage C is connected to the static nozzle S and gage A to the dynamic nozzle of a Pitot tube. The height of water H is directly proportional to the velocity of steam flowing through pipe P and automatically adjusts itself to the variations in velocity; thus, for decreasing velocities the water in glass G discharges itself through tube D until the water column H balances the velocity pressure in pipe P, and for increasing velocities condensation from the upper part of the instrument accumulates and the water column H rises until a balance is effected for the higher velocity.

9 The velocity of flow is determined by the well known equation:

$$V = c\sqrt{2gH} \tag{1}$$

in which

V = maximum velocity of flow, ft. per sec. Dynamic nozzle D is inserted in middle of pipe where the velocity is a maximum.

c = coefficient determined by experiment.

g = acceleration of gravity = 32.2.

H = height of a column of steam equal in weight to water column H.

Equation (1) may be expressed:

$$V_1 = 139\,c\sqrt{h\frac{d_w}{d_s}} \tag{2}$$

in which

V_1 = maximum velocity, ft. per min.

d_w = weight of 1 cu. ft. of water in gage glass G.

d_s = weight of 1 cu. ft. of the steam or mixture in pipe P.

h = height of column H in inches.

The weight of steam flowing per hour may be determined by substituting the proper quantities in equation (2), thus:

$$W = 58\,acr\sqrt{h\,d_s d_w} \tag{3}$$

in which

W = weight of steam flowing, lb. per hr.

a = area of the pipe, sq. in.; other notations as in (2).

r = ratio of the mean velocity to the maximum.

10 Equations (2) and (3) are general and are applicable to any size pipe and any pressure and quality of steam. For a given size of pipe, say 3 in. (extra heavy), and a given pressure of steam, say 70 lb. gage, equations (2) and (3) assume the following simple forms:

$$V = 2435\,cr\sqrt{h} \tag{4}$$

$$W = 1292\,cr\sqrt{h} \tag{5}$$

11 Tests with pipes 1 to 6 in. in diameter gave r a value of 0.79 to 0.84. For a 3 in. extra heavy pipe this value was 0.82 and remained practically constant for all velocities and pressures (atmospheric to 100 lb. gage.) Substitute this value of r in (4) and (5),

$$v = 1996\ c_1 \sqrt{h} \tag{6}$$

$$w = 1060\ c_1 \sqrt{h} \tag{7}$$

in which

v = actual mean velocity, ft. per min.
w = actual weight of steam flowing, lb. per hr.
c_1 = a coefficient determined by experiment.

Coefficient c_1 varied from 0.8 to 1.2 for the simple instrument in Fig. 1.

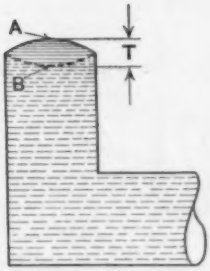

FIG. 3 INFLUENCE OF SURFACE TENSION ON HEIGHT OF WATER IN DYNAMIC NOZZLE

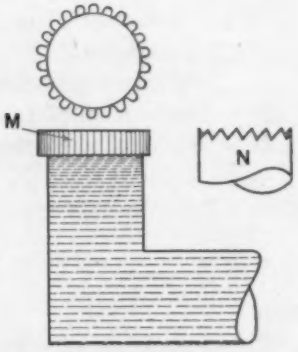

FIG. 4 DYNAMIC NOZZLE WITH ARRANGEMENT FOR MAINTAINING A CONSTANT LEVEL OF OVERFLOW

12 Variation in the value of c_1 was found to be due to

a Fluctuation in height T (Fig. 3), at the end of the dynamic tube due to surface tension. With a plain tube this variation amounted to as much as 0.25 in.
b Variation in density of water column.
c Capillarity in gage glass G.
d Aspiration in static tube at high velocities.

13 (*a*) A number of devices were constructed for eliminating the variation in height of T in the dynamic nozzle but all proved inefficient, except that shown in Fig. 4. By serrating the tube as indicated in N, Fig. 4, and surrounding it with a corrugated ferrule M, thereby forming a series of capillary tubes, the variation was practically eliminated, amounting to but 0.02 in.

14 (*b*) The variation in density of the water column is an inherent defect of this type of meter and cannot be remedied in this simple form of apparatus. The experiments gave a range in temperature

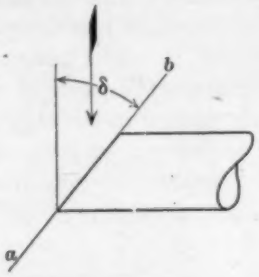

FIG. 5 STATIC NOZZLE CORRECTED FOR ASPIRATION

of 150 to 300 deg. fahr. resulting in a maximum possible error of 6 per cent. For high velocities the fluctuation in temperature is negligible but for low velocities the range may be considerable.

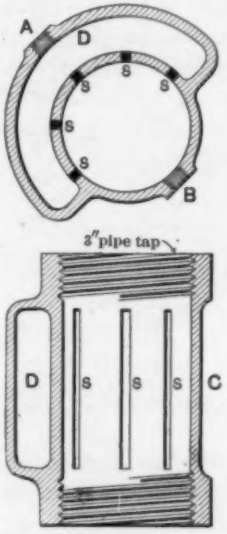

FIG. 6 DEVICE FOR DETERMINING ASPIRATION EFFECT

15 (*c*) Capillarity in the gage glass increases as the diameter of the glass decreases and may be considerable in tubes of small bore. With a $\frac{3}{4}$ in. tube it amounts to 0.05 in., hence its influence is negligible for high velocities.

16 (*d*) Aspiration in the static tube is appreciable only with velocities above 6000 ft. per min. It may be entirely eliminated by beveling the tube as indicated in Fig. 5. In this device the dynamic and aspiration effects neutralize each other and only the true pressure is recorded. The aspiration effect was determined by means of the apparatus illustrated in Fig. 6. It consists of a special fitting containing a chamber in communication with the main pipe, but so constructed that the velocity of flow in the chamber is practically eliminated. The difference in pressure between the two openings *A* and *B* is due to aspiration. Further experiments are necessary to show whether any fixed angle is applicable to all velocities.

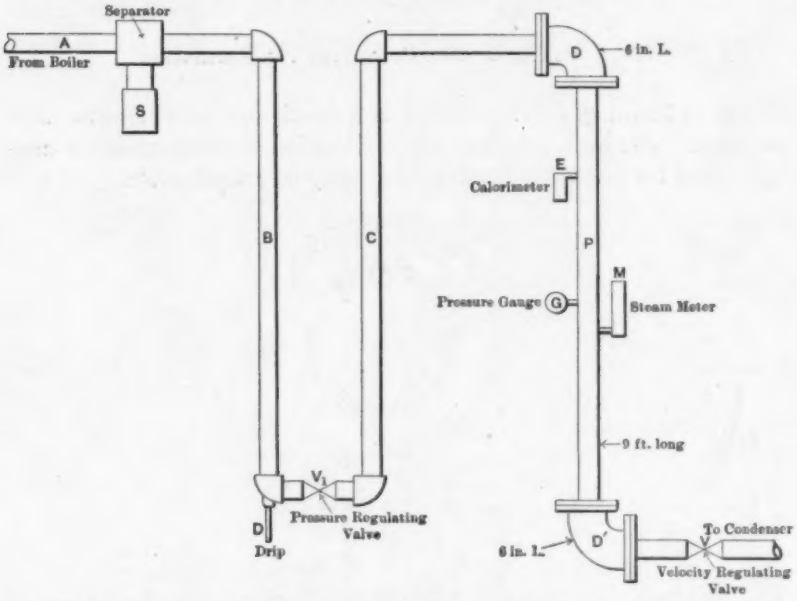

FIG. 7 DIAGRAMMATIC ARRANGEMENT OF PIPING FOR TESTING STEAM METERS

17 A simple construction, as illustrated in Fig. 2, with $\frac{1}{2}$ in. pipe connections and $\frac{3}{4}$ in. gage glass, fitted with dynamic tube, as illustrated in Fig. 4, and static tube as in Fig. 5, is an accurate means of indicating the true velocity of flow and the actual weight of water discharged for all velocities above that corresponding to $1\frac{1}{2}$ in. of water. For lower velocities any error in reading is so large in proportion to the entire head as to make considerable difference in results. The scale may be graduated to read velocities in feet per minute and the water rate in pounds per hour.

TABLE 1 TEST OF SIMPLE GAGE GLASS METER

STEAM PRESSURE 70-LB. GAGE. STEAM DRY. 3-IN. EXTRA HEAVY PIPE.

h	VELOCITIES FEET PER MINUTE			WEIGHTS POUNDS PER HOUR			
	ACTUAL MEAN	MAXIMUM BY METER	RATIO	ACTUAL	METER	DIFFERENCE	ERROR
2¼	2860	3528	1.23	1516	1545	29	1.88
3¾	3890	4685	1.20	2062	2052	10	0.48
5¼	4530	5480	1.21	2400	2400	0	0.00
7¼	5360	6570	1.22	2852	2878	26	0.91
9¾	6170	7405	1.20	3270	3240	30	0.92
		Average	1.21				

Mean Radius 0.82. Coefficient of Meter = 0.82 × 1.21 = 0.992 or practically unity.
Equation for meter: $V = 1996\sqrt{h}$, and $W = 1060\sqrt{h}$.

18 The results of a few tests of this simple device are given in Table 1, and the arrangements of piping for conducting the tests is shown diagrammatically in Fig. 7. The results are also plotted in Fig. 8. The curves give the weights as calculated from equation (7) and the small circles the weights as determined from the condensed steam. It will be noted that coefficient c_1 is unity and no calibra-

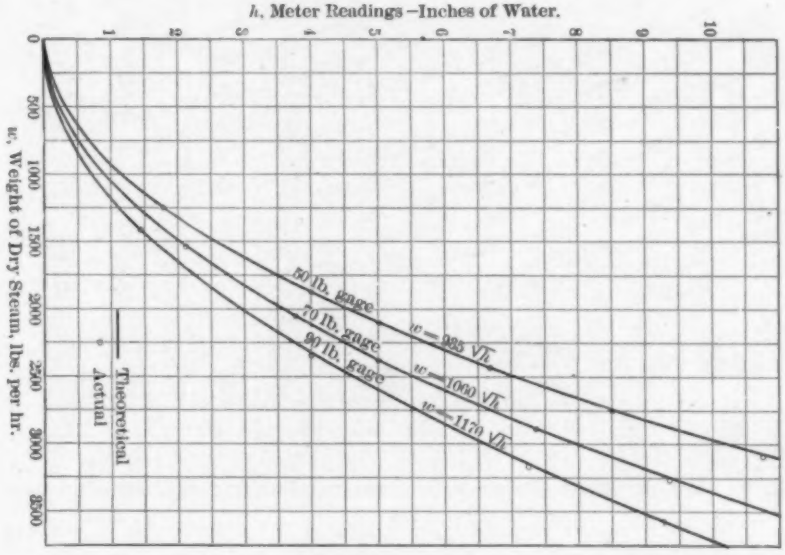

FIG. 8 TEST OF SIMPLE GAGE-GLASS METER, 3-IN. EXTRA HEAVY PIPE

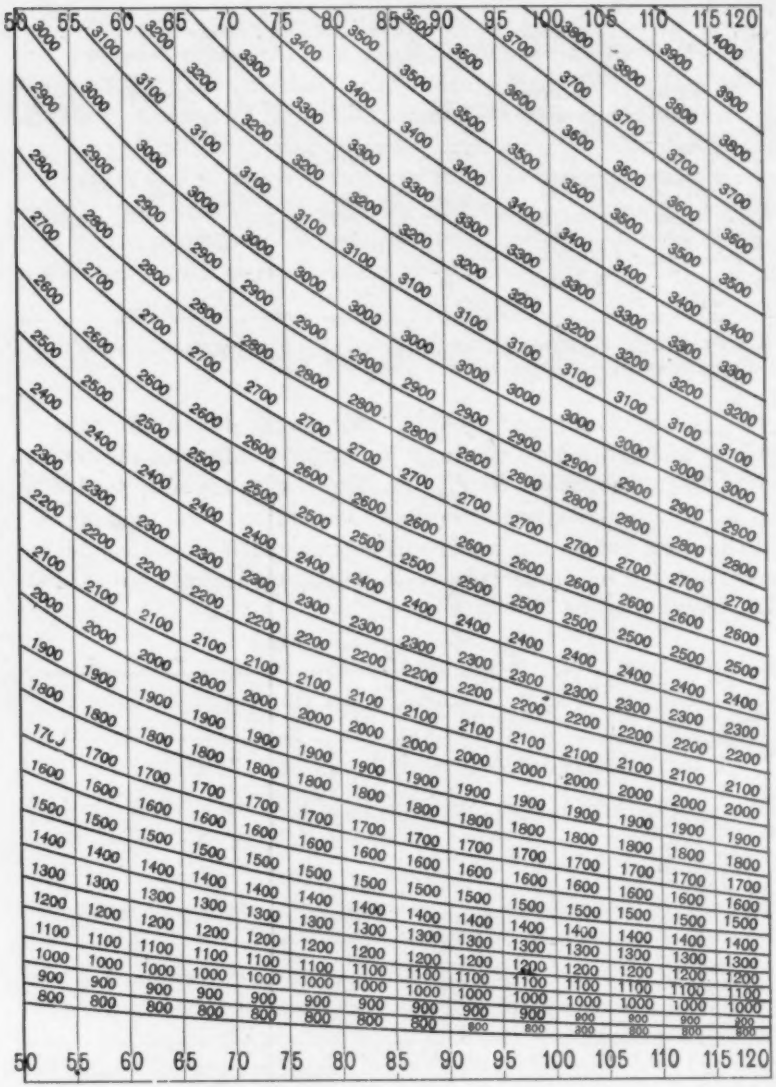

FIG. 9 TYPE OF CHART GIVING WIDE PRESSURE RANGE FOR GIVEN SIZE OF PIPE

tion is necessary. Tests on 1-in., 2-in., 3-in., and 4-in. standard pipe coefficient of practically unity. The tests tend to show that for a given size of pipe and a constant pressure and quality of steam

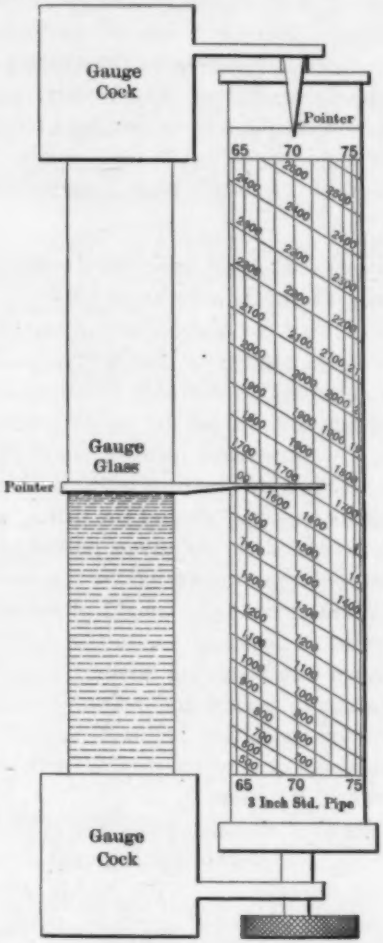

FIG. 10 METHOD OF MOUNTING CHART

the actual mean velocity and weight of water may be accurately determined by equations:

$$v = 139 \sqrt{h \frac{d_w}{d_s}} \tag{8}$$

$$w = 58a \sqrt{h d_w d_s} \tag{9}$$

This is strictly true for continuous flow only. Interrupted flow, as in connection with reciprocating engines, creates a fluctuating water column and it is difficult to obtain the mean readings. For engines making over 100 strokes per minute the height of water is practically constant but for lower speeds the fluctuation increases with the decrease in speed. By suitably throttling the lower valve V, water column H may be made to assume a fairly approximate mean value for speeds as low as 20 strokes per minute in engines taking steam full stroke.

19 The limitations of the simple gage glass meter for commercial purposes are:

a It is purely an indicating device and readings must be taken frequently to obtain average results.

b A scale graduated for a given set of conditions is accurate only for these conditions, the degree of accuracy varying with the fluctuation in steam pressure, change in quality of the steam and variations in temperature of the water column. A convenient form of chart for a given size of pipe and for a wide pressure range is illustrated in Fig. 9. The chart is wrapped around the drum, as in Fig. 10, and set to correspond to the given pressure. The height of the water column transferred to the chart by a suitable pointer gives at once the weight of steam flowing.

c Rapid increase in flow may cause the water in gage glass G to be blown out requiring several minutes for sufficient condensation to collect and balance the velocity head.

d Inaccuracy for velocities below that corresponding to a 1½ in. water column, or roughly, 2000 ft. per min. for pressures over 70 lb. gage.

e Cannot be used for measuring the flow of highly superheated steam except for practically constant flow and constant degree of superheat. With highly superheated steam a condensation chamber must be fitted to upper gage cock C, Fig. 2.

20 Fig. 11 shows a modification of the simple gage glass meter with simple pipe connection, which requires the pipe line to be tapped in only one place. It may be set at any angle with the horizontal, thereby increasing the sensitiveness of the readings.

21 Fig. 12 shows an application of the gage glass meter in which many of the defects of the simple gage glass device are eliminated.

The error due to the variation in water level at the end of the dynamic nozzle is entirely eliminated by making the tube a "dry" tube; i. e., water of condensation is not returned through the dynamic nozzle. The temperature of the water in the gage glass is practically constant, all condensation from the static end being discharged through drain F to the chamber below, and all water carried over by the dynamic end being discharged into pipe P directly to overflow G. The area and volume of chamber M is so large compared with that of glass W that sudden variations in flow do not materially affect the level of

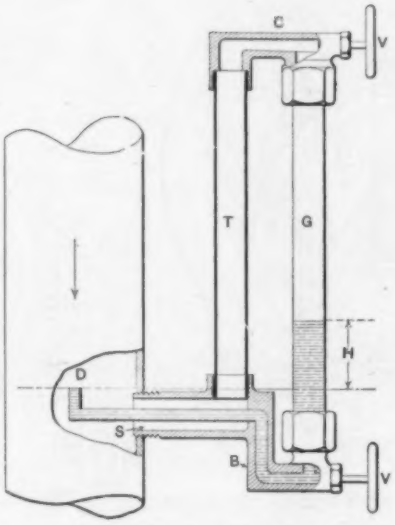

FIG. 11 GAGE-GLASS METER WITH SINGLE PIPE CONNECTION

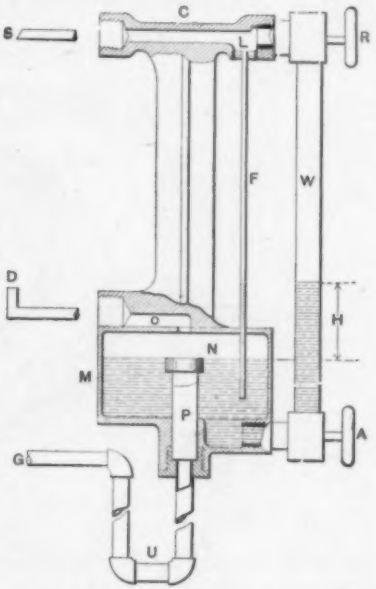

FIG. 12 GAGE-GLASS METER WITH DRY DYNAMIC TUBE

water in M and cannot blow the water out of glass W. The only defect in the instrument is the error due to capillarity in glass G, which, as stated before, amounts to but 0.05 in. for a $\frac{3}{4}$ in. tube.

22 The operation is as follows: Velocity pressure is transmitted through tube D and opening O, into the body of chamber M. This pressure acting on the surface of the condensation in the chamber forces water into glass W until a balance is effected. Condensation is discharged continuously through pipe P and water seal U of the main pipe. Tests of this meter gave practically theoretical results

for all velocities ranging from the equivalent of a ½-in. water column to 10-in.

23 Fig. 13 shows an application of the self-adjusting water column for very low velocities, 2000 ft. per min. and under. Glass ball *B* rises and falls with the water column, transmitting its motion through levers *L* and *N*, gear sector *G* and pinion *P*, to permanent magnet, *M*. The latter transmits its motion to magnet *M′* outside the casing. Pointer *C* is fastened to magnet *M′*. Thus any motion of ball *B* is multiplied by pointer *C* and indicated on dial *K*. Magnets *M* and

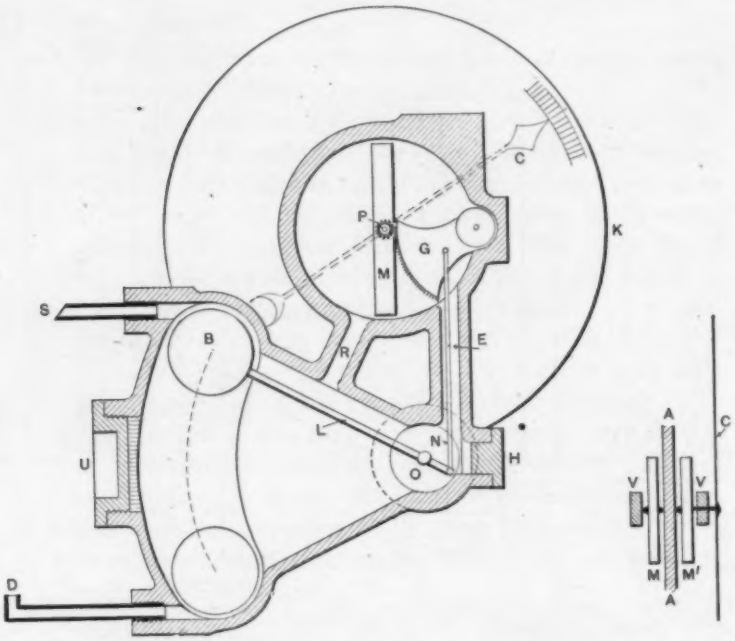

FIG. 13 METER WITH SELF-ADJUSTING WATER COLUMN FOR LOW VELOCITIES

M′ are independently mounted on pivot bearings, *M* within the casing and subjected to steam temperature, the other outside the casing. Motion of *M* is transmitted magnetically to *M′* through the casing, thus doing away with stuffing boxes. The relative positions of magnets and casing are illustrated in the lower corner of Fig. 13. On account of the angularity of the connecting links and the frictional resistances, small as they are, the dial graduations cannot be conveniently calculated but must be calibrated by experiment.

24 This instrument is very sensitive, indicating velocity changes of 20 ft. per min. When connected directly to the pipe it is subject to all of the errors of the simple gage glass meter and is altogether too sensitive for accuracy. When connected, however, to the chamber illustrated in Fig. 12 taking the place of the gage glass, velocity changes as low as 20 ft. per min. have been accurately determined.

25 Only a few experiments have been made with this device and further tests are necessary before definite statements can be made relative to its reliability.

26 All of the devices described above are simple indicating mechanisms, and with the exception of the one illustrated in Fig. 13, cannot conveniently be made autographic.

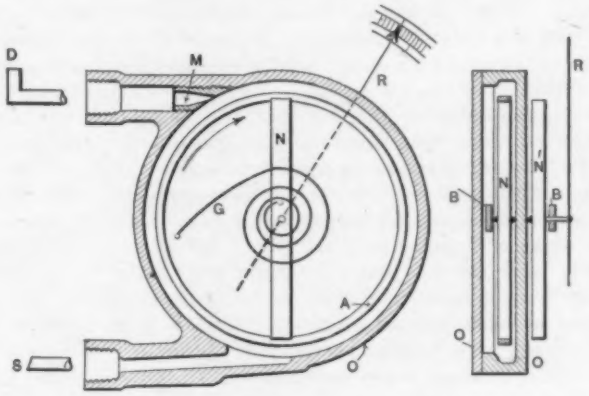

FIG. 14 INDICATING IMPULSE METER

27 For commercial purposes a steam meter should be autographic, or better still, integrating. The ideal meter is one which shows at a glance the weight of steam flowing for any given period of time and which may be read as one does a watt-hour meter.

28 Fig. 14 shows an application of the Pitot tube for indicating, autographically, the weight of steam discharged and differing basically from those just described. A permanent magnet N forms the spoke of a small aluminum wheel A. Rotation of wheel A is resisted by spiral spring G. D and S are dynamic and static nozzles, respectively, of a Pitot tube. The velocity head discharges a small jet of steam through nozzle M and exerts a force on the periphery of wheel A, tending to rotate it about its axis. The angular rotation increases with the velocity. The motion of wheel A is imparted through the

medium of magnets N and N' to pointer R. By means of a suitable clock-work the angular movement of wheel A may be autographically transferred to a chart giving a continuous record of the weight of steam flowing.

29 By permitting the wheel to rotate and by connecting magnet N' to a series of rotary dials an integrating or total output mechanism is readily effected.

30 Experiments are now being conducted with the autographic and integrating devices just described, but sufficient data are not yet available as to their respective merits.

COMMENT ON CURRENT BOOKS

CONCRETE, PLAIN AND REINFORCED. By Frederick W. Taylor and Sanford E. Thompson. *John Wiley & Sons, New York.* Second Edition. 1909. Cloth, 6 by 9; xi + 807 p.; 249 illustrations. Price $5.

This second edition of the book brings the treatment of the subject up to date, more than 200 pages of text and tables having been added. The new chapter one outlines the essentials of concrete construction and the errors frequently made, with references to pages in which more detailed information is given. The specifications for cement and concrete in chapter iii, and chapter ix by W. B. Fuller on proportioning, have been revised. Chapters xiv and xv on mixing and depositing have been enlarged, giving more recent tests on the strength and permeability of concrete. Chapter xxi on reinforced concrete design has been increased from 51 to 131 pages. Tables and diagrams for use in designing cover twenty pages, and a complete example of floor design gives the mathematical computations in detail. Prof. F. P. McKibben has written the chapter on arches, discussing the design of the arch by the elastic theory, a complete example being given with all the steps in the calculations. Chapter xxix is a brief history of the development of cement manufacture, with an outline of modern processes. The list of references to concrete literature in chapter xxxi has been increased over 50 per cent. In order to simplify the formula used, the demonstrations have been placed as far as possible in footnotes and appendices.

Contents by chapter headings: Essential Elements in Concrete Construction; Concrete Data; Elementary Outline of the Process of Concreting; Specifications; The Choice of Cement; Classifications of Cement; Chemistry of Hydraulic Cements, by Spencer B. Newberry; Standard Cement Tests; Speecial Tests of Cement and Mortar; Strength and Composition of Cement Mortars; Voids and other Characteristics of Concrete Aggregates; Proportioning Concrete, by William B. Fuller; Tables of Quantities of Materials for Concrete and Mortar; Preparations of Materials for Concrete; Mixing Concrete; Depositing Concrete; Effect of Sea Water upon Concrete and Mortar by R. Feret; Laying Concrete and Mortar in Freezing Weather; Fire and Rust Protection; Water Tightness; Strength of Plain Concrete; Reinforced Concrete Design; Arches by Frank P. McKibben; Sidewalks, Basement Floors and Pavements; Concrete Building Construction; Foundations and Piers; Dams and Retaining Walls; Conduits and Tunnels; Reservoirs and Tanks; Cement Manufacture; Miscellaneous Structures; References to Concrete Literature; Appendices.

CONCRETE. By John C. Trautwine, Jr., and John C. Trautwine, 3d. *John Wiley & Sons, New York.* 1909. Cloth, 5 by 7; ix + 190 p. Price $2.

The book is essentially the section on concrete in the latest edition of the authors' Pocket-Book, with the addition of chapters on strength of materials, mortar and sand—also from the pocket-book. Besides dealing very fully with

the characteristics of cement and concrete, the book gives valuable and extensive data regarding the practice of the art,—a feature mentioned in the review of the Pocket-Book below.

Contents by chapter headings: Strength of Materials; Cement Mortar; Aggregates; Reinforced Concrete; Experiment and Practice; Digest of Specifications; Cost.

CIVIL ENGINEER'S POCKET-BOOK. Revised by John C. Trautwine, Jr., and John C. Trautwine, 3d. *John Wiley & Sons, New York.* 1909. Nineteenth Edition. Leather, pocket size; xxxii + 1257 p. Price $5.

One of the prominent features of the latest edition of this well-known book is the section on cement and concrete, increased from 17 pages in the 1902 edition to 126 pages in the present one. The section has its own index for ready reference to the "Selected Results of Experiment and Practice;" these data, with the "Digest of Specifications" for general concrete work, make the book an extremely valuable source of information. Revisions of the pages on strength of materials; columns; specifications for steel rails, rail joints, iron and steel; logarithmic trigonometric functions; hydraulics; price list; business directory; and bibliography, have helped to add 300 new pages to the book.

DIGEST OF DATA COLLECTED BEFORE THE YEAR 1908 RELATING TO THE SANITARY CONDITION OF NEW YORK HARBOR. *Martin B. Brown Press, New York.* 1909.

The Metropolitan Sewerage Commission of New York, of which H. deB. Parsons, Mem.Am.Soc.M.E., is a member, has prepared and issued this report for the purpose of making public a digest of the analytical data available up to 1908 relating to the sanitary condition of the waters of New York harbor. The investigations here included contribute materially to an understanding of the conditions which follow the discharge of sewage into harbor waters, but are not presented as a basis for conclusions, further examinations of the water of New York harbor and the procurement of additional data being now in progress.

BETTERMENT BRIEFS. A Collection of Published Papers on Organized Industrial Efficiency. By H. W. Jacobs. *John Wiley & Sons, New York, 1909.* Cloth, 6 by 9; 240 pages; 136 illustrations. Price $3.50.

This book, as its sub-title indicates, is a collection of papers by the author on the "movement for the betterment of American railroading." The subjects treated are varied, covering the field from shop methods to welfare work for the employees. Though the author writes from the viewpoint of the railroad shop, much that he says may be applied to other fields of industrial activity. The book with its numerous illustrations should make interesting reading; the only criticism that may be made is of the reddish brown tint of the ink used for the text.

Contents by chapter headings: Commercial Tool Methods in Railroad Shops; Improved Devices for Railroad Shops; High-Speed Steel in Railroad Shops; Practical Advice to College Men; Organization and Efficiency in the Railway Machine Shop; The Relation Between the Mechanical and Store Departments; Shop Efficiency; The Square Deal to the Railway Employee.

ROBERT FULTON AND THE CLERMONT. By Alice Crary Sutcliffe. *The Century Co., New York*, 1909. Cloth, 5 by 7½, xv + 367 pages; illustrated. Price $1.20.

The author, Fulton's great-granddaughter, has given an interesting account of Fulton's early life, his work in steamboat navigation leading up to the building of the Clermont, with some description of his work on canals, torpedoes, and submarines. Doubtless this was the author's intention, rather than to reply to the charges that Fulton was a plagiarist and incapable himself of developing the steamboat without the aid of earlier or contemporary investigators. Though the reader may obtain the impression that Fulton would never have built the Clermont without the backing of Livingston, yet in all his work he showed such an indefatigable spirit that it may be assumed that he would have gone far toward attaining his end by some other means. Of the whole matter it may be said that Fulton should not be celebrated for accomplishing a great engineering feat, but rather for making, by whatever means, the first successful commercial application of steam power to navigation. The author has evidently spent much time and care on the book, and many interesting letters and drawings are reproduced.

Contents by chapter headings: Early Life; Robert Fulton in France; The Trial Boat on the Seine; The Clermont; Appendix.

BIBLIOGRAPHY OF THE COTTON MANUFACTURE. By C. J. H. Woodbury, A.M. Sc.D. *Published by the author at Boston, Mass.* 1909. Cloth 7 by 10 213 p. Price $2.

The author's work as secretary of the National Association of Cotton Manufacturers has given him a valuable viewpoint from which to select the important literature of the art. Information has been drawn not only from the manufacturers, but from leading libraries on both sides of the Atlantic, making the list very complete as to works in English, French and German. This work is of value to investigators in the subjects of cotton manufacture, the finishing of goods, or the engineering and mechanical problems pertaining to the matter. It also affords a means of directing examinations into the agricultural, commercial and historical questions relating to cotton. Apart from the cotton manufacture in a technical sense, there are a large number of references to its history and economics bearing upon sociological problems involved in the relations of labor and capital.

Contents: Cotton manufacture; Finishing; Engineering and Machinery; History and Economics; Cotton; Journals.

ALUMINUM HANDBOOK. *Published by The Aluminum Company of America, Pittsburg, Pa.* Size 4½ by 7; 154 p.; illustrated.

Much interesting and valuable information regarding aluminum will be found in this book, which is made up of five pamphlets dealing with the properties of the metal, and its various products. It is attractively bound in leather, with a thumb-index, supplemented by a table of contents for each section. The typographical work is excellent and the illustrations are of a high order.

Contents: Properties of Aluminum; Alloys of Aluminum; Methods of Working Aluminum; Fabricated Aluminum; Useful Tables.

ACCESSIONS TO THE LIBRARY

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS. *Transactions.* Vol. 30. *New York, 1909.*

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EXCHANGES

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LEAD AND ZINC ORES OF SOUTH-WESTERN VIRGINIA. By M. M. Caldwell. *Morgantown, 1909.*

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UNITED ENGINEERING SOCIETY

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TRADE CATALOGUES

ALUMINUM COMPANY OF AMERICA, *Pittsburg, Pa.* Five pamphlets.

ARMSTRONG CORK CO., *Pittsburg, Pa.* Nonpareil Corkboard Insulation. 1909.

H. W. CALDWELL & SONS CO., *Chicago, Ill.* Catalogue No. 34; elevating, conveying, and power transmitting machinery. 847 pp.

DE LA VERGNE MACHINE CO., *New York City.* Oil engines, horizontal, four-cycle, and vertical two-cycle types. 84 pp.

GENERAL ELECTRIC CO., *Schenectady, N. Y.* Bulletin No. 4691. Small polyphase motors, 6 pp.; No. 4694. Portable sub-stations for electric railways, 8 pp.; No. 4695. G-I type K enclosed arc lamps, 16 pp.; No. 4698. Luminous arc headlights, 6 pp.; No. 4699. Motor-driven air compressors, geared type, 6 pp.; No. 4700. Switchboard instruments, type R-2, 5 pp.; No. 4701. Emergency straight air brake system, 10 pp.; No. 4704. G. E.-219 railway motor, 15 pp. Price list No. 5210, 8 pp. Folder on contents of November G. E. Review, 3 pp.

GLIDDEN VARNISH CO., *Cleveland, O.* Protection of iron and steel against corrosion by acid, and graphite acid-proof coating. 8 pp. Advanced finishes for modern building construction. 8 pp.

GOLDEN-ANDERSON VALVE SPECIALTY CO., *Pittsburg, Pa.* Catalogue No. 12. Valves, steam and water specialties. 68 pp.

LANGDON-DAVIES MOTOR CO., LTD., *London.* Size and types of motors, also useful tables. 8 pp.

SANDUSKY FOUNDRY & MACHINE CO., *Sandusky, O.* Paper made by "Millspaugh" suction roll process. 20 pp.

UNDERFEED STOKER CO. OF AMERICA. *Chicago, Ill.* Publicity magazine, October 1909. 15 pp.

UNITED ELECTRIC LIGHT & POWER CO., *New York City.* Light, September 1909. 12 pp.

L. J. WING MANUFACTURING CO., *New York City.* Bulletin No. 6. "Typhoon" turbine blower for mechanical draft. 15 pp.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society, and these are on file, with the names of other good men not members of the Society who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

080 Experienced draftsman in general rolling-mill work, especially new construction, at a large steel plant in the middle West. State experience and salary expected.

081 Position competent designer of mathematical and optical instruments at Frankford Arsenal, Philadelphia, Pa. Address the Commanding Officer, giving full particulars as to training and experience.

082 Wanted—A thoroughly practical and energetic mechanic to take position as rate fixer on premium work in machine shop building a line of air compressors. Applicant must have had broad experience in practical shop work.

083 Assistant professorship, in charge of design courses in engines, steam turbines, locomotives or gas engines, with assured advancement to full professorship in few years, for the right man. Institution desirous of having its men do outside work. Want a man of ability and experience. Position would pay initially from $1800 to $2000. Location, New York state.

084 Thouroughly competent young man to take a position as assistant engineer, particularly in connection with automobile work. Would like to get in touch with a man who has had shop experience, particularly in connection with heat treatments of steels and other metals.

MEN AVAILABLE

321 Superintendent and manager desires change for larger opportunity. High grade organizer and executive. Specialized on equipment, production, and costs.

322 General manager or assistant, graduate M.E., at present holding similar position, would like to make change; ten years practical experience; good executive ability, best of references.

323 Technical graduate desires position as assistant to superintendent, testing engineer, or similar position; age thirty-three; has employed and had charge of men.

324 Junior, experienced in power-plant design and installation; manufacturing of condensers, pumps, piping, etc.; shop and business experience; desires responsible position, best of references.

325 Member, forty-three years of age, graduate M.E., experience as draftsman, chief draftsman, mechanical engineer, and chief designer of steam engines and compressors. Will accept position after February 1, 1910, with firm building or contemplating building this class of machinery.

326 Junior member, technical graduate, mechanical, six years experience, at present employed. Designing, estimating costs, inspecting and erecting with car-building company; assistant engineer with engineers and contractors on power and industrial plant construction. Would prefer connection from January 1, 1910, with engineering organization or manufacturer in New York city, as assistant engineer, assistant manager or sales engineer.

327 Member would like connection with firm, or individual manufacturer who is looking forward to gradually retiring from active oversight of business and desires some one to assume the responsibility. Possess a well rounded experience in shop work and designing, and as general superintendent covering all sides of a business. Personal interview preferred.

328 Junior member, graduate mechanical engineer, specialist on gas engines and refrigeration, would like to become associated with an established consulting engineer, or with a firm needing an up-to-date man.

CHANGES IN MEMBERSHIP

CHANGES OF ADDRESS

ADAMS, Kilburn E. (Junior, 1908), Mech. Engr., Boston & Albany R. R., Rm. 372, South Sta., Boston, and *for mail*, 123 Oxford St., North Cambridge, Mass.

ADAMS, Thomas D. (Junior, 1906), Werner & Pfleiderer, and *for mail*, 619 Emerson St., Saginaw, Mich.

ALEXANDER, Edward E. (Associate, 1908), Ch. Draftsman, Am. Loco. Co. Cooke Wks., and *for mail*, 830 E. 23d St., Paterson, N. J.

ANDERSON, Harry Warfield (Associate, 1907), Anderson-Kent Co., 1030 Candler Bldg., and *for mail*, Owens Apts., 5 E. 3d St., Atlanta, Ga.

ARD, Charles Edgar (1908), Mgr., Christopher, Ard & Co., Engrs. and Contrs., Starkville, Miss.

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BANTA, Earle J. (1907), Cincinnati Equipment Co., Cincinnati, O.

BARNES, Charles B. (1905; 1908), Mech. Engr., Holabird & Roche, Archts., Monadnock Bldg., and *for mail*, 1400 E. 53d St., Chicago, Ill.

BEECHER, J. F. (Associate, 1908), Draftsman, Pa. Steel Co., and *for mail*, 310 N. 2d St., Harrisburg, Pa.

BIGELOW, Charles H. (1904), with Chas. T. Main, Mill Engr., 45 Milk St., Boston, and *for mail*, 46 Calhaun Ave., Everett, Mass.

BLANCHARD, Arthur S. (Associate, 1909), Asst. Genl. Mgr., Birdsboro Steel Fdy. & Mch. Co., Birdsboro, Pa.

BRIDGE, James W. (Associate, 1905), Boswell, Pa.

BROWN, Robt. S. (1891; 1904), Secy., New Britain Mch. Co., and *for mail*, 16 Forest St., New Britain, Conn.

BURGESS, Edward W. (Junior, 1908), Metzger Motor Car Co., and *for mail*, 147 Milwaukee Ave. E., Detroit, Mich.

CHAMBERLAIN, Geo. E. (1907), Pres., Lewell Mfg. Co., 1402 Mich. Ave., Chicago, Ill.

CONLEE, George D. (Junior, 1906), Supt., Binghamton Gas Wks., 40 Chenango St., and *for mail*, 71 Carroll St., Binghamton, N. Y.

COX, Claude E. (Associate, 1907), H. E. Wilcox Motor Car Co., Minneapolis, Minn.

CRANE, William Edward (1887), Broadalbin, N. Y.

CRAWLEY, George E. (Junior, 1908), 548 W. 124th St., New York, N. Y.

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DAVIS, Edwin Halsted (Junior, 1907), Ch. Draftsman, Davidson's Steam Pump Wks., and *for mail*, 256 Martense St., Brooklyn, N. Y.

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FORSYTH, William (1883), Manager, 1888–1891; Mech. Engr., R.R. Age Gazette, 303 Dearborn St., and Windermere Hotel, Chicago, Ill.

FRITZ, Aime L. G. (Junior, 1907), Tee Square and Triangle Co., 30–32 Clinton St., Newark, N. J.

FROHWEIN, Richard W. (Junior, 1907), Pur. Agt., M. H. Treadwell Co., 140 Cedar St., New York, N. Y., and *for mail*, 431 Magee St., Elizabeth, N. J.

GARDNER, Horace C. (1904), Mgr. Constr. and Mech. Depts., Swift & Co., and *for mail*, Kenwood Hotel, Chicago, Ill.

GERRISH, William H. (1901), Supt. Soft Fibre Dept., Columbian Rope Co., and *for mail*, 164 W. Genesee St., Auburn, N. Y.

GIELE, Walter S. (Junior, 1908), Harrison Safety Boiler Wks., Philadelphia, Pa., and *for mail*, 3 Hamilton Park, New Brighton, S. I., N. Y.

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GODDARD, Arthur L. (1903), 1618 School St., Rockford, Ill.

GOWIE, William (1905), Emily & Warren Aves., Crafton, Pa.

GREEN, Chas. Henry (Junior, 1905), Member of Firm, M. A. Earl & Co., Carthage, Mo.

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NEW MEMBERS

JACKSON, F. W. (1909), Dist. Mgr., Harrisburg Fdy. & Mch. Wks., Continental Trust Bldg., Baltimore, Md.

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GAS POWER SECTION

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SAGE, Darrow (Affiliate, 1908), Ch. Engr., J. C. P. H., Hudson & Manhattan R. R. Co., Jersey City and *for mail*, 20 Ivanhoe Terrace, East Orange N. J.

SAMPSON, Charles C. (Affiliate, 1909), Inspr., Gas Power Mchy., Ill. Steel Co., South Chicago, and *for mail*, 646 E. 73d St., Chicago, Ill.

STUDENT SECTION

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SHULTS, L. J. (Student, 1909), 1860 S. Kedzie Ave., Chicago, Ill.
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MESTON, A. F. (Student, 1909), Stanford Univ., Cal.
NORWOOD, C. T. (Student, 1909), Stanford Univ., Cal.
STEINBECK, C. E. (Student, 1909), Kingman, Ariz.
WARREN, H. C. (Student, 1909), Stanford Univ., Cal.

UNIVERSITY OF ILLINOIS

DUNSHEATH, L. M. (Student, 1909), 509 S. 4th St., Champaign, Ill.
HEILMAN, H. C. (Student, 1909), 912 S. Nevada St., Urbana, Ill.
HUNTINGTON, C. S. (Student, 1909), 906 S. Fifth Ave., Champaign, Ill.
KREIDLER, D. W. (Student, 1909), 410 Chalmes St., Champaign, Ill.
KUPPATRICK, H. J. (Student, 1909), 501 E. Green St., Champaign, Ill.
LUND, J. (Student, 1909), 506 E. John St., Champaign, Ill.
SPONSEL, J. G. (Student, 1909), 304 E. Daniel St., Champaign, Ill.
WESTLAND, A. F. (Student, 1909), 1202 W. Main St., Urbana, Ill.

COMING MEETINGS

November and December

Secretaries or members of societies whose meetings are of interest to engineers are invited to send in their notices for publication in this department. Such notices should be in the editor's hands by the 18th of the month preceding the meeting.

ALABAMA LIGHT AND TRACTION ASSOCIATION
November 15, 16, annual convention, Birmingham. Secy., Lloyd Lyon, 158 Government St., Mobile.

AMERICAN ASSOCIATION FOR THE ADVANCEMENT OF SCIENCE
December 27, Boston, Mass. Secy., L. O. Howard, Smithsonian Institution, Washington, D. C.

AMERICAN CIVIC ASSOCIATION
November 15–19, Cincinnati, O. Secy., Richard B. Watrous, Harrisburg, Pa.

AMERICAN FEDERATION OF TEACHERS OF MATHEMATICS
December 28, 29, annual meeting, Baltimore, Md. Secy., C. R. Mann, University of Chicago.

AMERICAN INSTITUTE OF ARCHITECTS
December 14–16, annual convention, Washington, D. C. Secy., Glenn Brown, Octagon Bldg.

AMERICAN INSTITUTE OF CHEMICAL ENGINEERS
December 8–10, annual meeting, Philadelphia, Pa. Secy., J. C. Olsen, Polytechnic Institute, Brooklyn, N. Y.

AMERICAN MATHEMATICAL SOCIETY
November 27, University of Missouri, Columbia, Mo., Southwestern Section. Secy., O. D. Kellogg, 411 Hitt St.

AMERICAN PHYSICAL SOCIETY
November 27, University of Illinois, Urbana, Ill. Secy., Ernest Merritt, Ithaca, N. Y.

AMERICAN RAILWAY ASSOCIATION
November 17, annual meeting, Chicago, Ill. Secy., W. F. Allen, 24 Park Pl., New York.

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
November 17, Boston, Mass. December 7–10, annual meeting, 29 W. 39th St., New York. July 26–29, 1910, joint meeting with Institution of Mechanical Engineers, Great Britain. Secy., Calvin W. Rice, New York.

AMERICAN SOCIETY OF AGRICULTURAL ENGINEERS
December 28, 29, annual meeting, Ames, Ia. Secy., L. W. Chase, Univ. of Neb., Lincoln, Neb.

AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
December 6, New York meeting. Secy. W. H. Ross, 154 Nassau St.

AMERICAN SOCIETY OF SWEDISH ENGINEERS

November 20, 271 Hicks St., Brooklyn, N, Y. Paper: Magnetic Separation of Iron Ores, N. V. Hansell. Secy., E. Hammerstrom.

ASSOCIATION OF AMERICAN PORTLAND CEMENT MANUACTURERS

December 14, 15, annual meeting, New York. Secy., P. H. Wilson, Land Title Bldg., Philadelphia, Pa.

ASSOCIATION OF TRANSPORTATION AND CAR ACCOUNTING OFFICERS

December 14, 15, Chattanooga, Tenn. Secy., G. P. Conard, 24 Park Pl., New York.

BROOKLYN ENGINEERS' CLUB

November 18, December 2, 117 Remsen St., Brooklyn, N. Y. Papers: Pneumatic Foundations, T. K. Thomson, Mem.Am.Soc.M.E.; Steel Sheet Piling, A. R. Archer. Secy., Joseph Strachan.

CENTRAL ELECTRIC RAILWAY ASSOCIATION

November 18, Claypool Hotel, Indianapolis, Ind. Papers: A Centralized Testing Organization, J. G. Callan; Publicity, A. D. B. Van Zandt; The Claim Adjusters' Association, E. C. Carpenter; The Transportation and Traffic Association, J. B. Crawford. Secy., A. L. Neereamer.

CENTRAL RAILWAY AND ENGINEERING CLUB OF CANADA

November 16, December 21, Prince George Hotel, Toronto. Papers: Gas Engines, their Origin and Commercial Use, D. M. Henderson; Gas Manufacture, C. G. Herring. Secy., C. J. Worth, Union Sta.

COLORADO SCIENTIFIC SOCIETY

December 18, annual meeting, Denver. Secy., Dr. W. A. Johnston, 801 Symes Bldg.

EASTERN ICE ASSOCIATION

November 18–20, New York.

EMPIRE STATE GAS AND ELECTRIC ASSOCIATION

November 17, 18, 29 W. 39th St., New York. Secy., C. H. B. Chapin.

ENGINEERS' CLUB OF PHILADELPHIA

November 20. Paper: Rating of Pitot Tubes for Use in Testing a Niagara Power Plant. Secy., W. P. Taylor.

ENGINEERS' CLUB OF ST. LOUIS

December 1, annual convention, 3817 Olive St. Secy., A. S. Langdorf, Washington University.

ENGINEERS' CLUB OF TORONTO

November 18. Paper: Facts and Figures Relating to Producer Plant Practice, M. Chapman. Secy., R. B. Wolsey.

FRANKLIN INSTITUTE

December 10, Witherspoon Hall, Philadelphia, Pa. Lecture: Perils of Peace, or A Safer America, W. H. Tolman.

MODERN SCIENCE CLUB

November 16, 23, 30, 125 S. Elliott Pl., Brooklyn, N. Y. Papers: Globe, Gate and Check Valves, Russell Bonnell; The Use of Incandescent Lamps in Illumination, B. F. Fisher, Jr.; Water Storage for Power in the Adirondacks, W. T. Donnelly, Mem.Am.Soc.M.E. Secy., J. A. Donnelly.

NATIONAL ASSOCIATION OF RAILWAY COMMISSIONERS

November 16, annnal meeting, Washington, D. C. Secy., M. S. Decker, Albany, N. Y.

NATIONAL CIVIC FEDERATION

November 22, 23, annual meeting, New York. Secy., D. L. Cease, Metropolitan Bldg.

NATIONAL COMMERCIAL GAS ASSOCIATION

December 12, 14, annual convention, Madison Square Garden, New York. Secy., L. S. Bigelow, Light Publishing Co., Willimantic, Conn.

NATIONAL GAS AND GASOLENE ENGINE ASSOCIATION

November 30, December 1, 2, LaSalle Hotel, Chicago, Ill. Secy., Albert Stritmatter, Cincinnati, O.

NATIONAL MUNICIPAL LEAGUE

November 15–19, Cincinnati, O. Secy., C. R. Woodruff, 121 S. Broad St., Philadelphia, Pa.

NATIONAL SOCIETY FOR PROMOTION OF INDUSTRIAL EDUCATION

December 1–3, annual convention, Milwaukee, Wis. Secy., J. C. Monaghan, 20 W. 44th St., New York.

NEW JERSEY SANITARY ASSOCIATION

December 3, 4, annual meeting, Laurel-in-the-Pines, Lakewood. Secy., J. A. Exton, 75 Beech St., Arlington.

NEW YORK RAILROAD CLUB

November 19, 29 W. 39th St. Paper: The Handling of Freight at Terminals, H. McI. Harding. Secy., H. D. Vought, 95 Liberty St.

OHIO SOCIETY MECHANICAL, ELECTRICAL AND STEAM ENGINEERS

November 19, 20, main annual meeting, Lima, O. Secy., David Gaehr, Schofield Bldg., Cleveland.

RICHMOND RAILROAD CLUB

December 13. Paper: Block Signals, Chas. Stephens. Secy., F. O. Robinson.

ROCHESTER ENGINEERING SOCIETY

December 10, annual meeting. Secy., John F. Skinner , 54 City Hall.

SHORT LINE RAILROAD ASSOCIATION

December 14, annual meeting, New York. Secy., J. N. Drake, 60 Wall St.

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

November 18–19, general meeting, 29 West 39th St., New York. Papers: Applications of Electricity to the Propulsion of Naval Vessels, W. L. R. Emmet, Mem.Am.Soc.M.E.; The Producer-Gas Boat Marenging, H. L. Aldrich, Assoc.Mem.Am.Soc.M.E.; The Foreign Trade Merchant Marine, G. W. Dickie, Mem.Am.Soc.M.E.; Material-Handling Arrangements for Vessels on the Great Lakes, A. E. Brown, Mem.Am.Soc.M.E., etc. Secy., W. J. Baxter.

SOUTHERN AND SOUTHWESTERN RAILWAY CLUB

November 18, annual meeting, Candler Bldg., Atlanta, Ga. Papers on Oil Lamps, Front-End Arrangements, Draft-Rigging. Secy., A. J. Merrill, 218 Prudential Bldg.

SOUTHWESTERN ICE MANUFACTURERS' ASSOCIATION

November 16–18, San Antonio, Tex.

WASHINGTON SOCIETY OF ENGINEERS

November 24, anniversary celebration. Secy., Paul Bausch.

WESTERN SOCIETY OF ENGINEERS

November 20, December 1, 1735 Monadnock Blk., Chicago, Ill. Papers: The Panama Railroad, Ralph Budd; Compressed Air in Contract Work, M. W. Briseler; River and Harbor Improvements at Chicago and the Calumet, T. H. Rees.

MEETINGS TO BE HELD IN THE ENGINEERING BUILDING

Date November	Society	Secretary	Time
16	New York Telephone Society	T. H. Lawrence	8.00
17–18	Empire State Gas and Electric Asso	C. H. B. Chapin	All day
18–19	Naval Architects and Marine Engineers	W. J. Baxter	All day
19	New York Railroad Club	H. D. Vought	8.15
24	Municipal Engineers of City of New York	C. D. Pollock	8.15
December			
1	Wireless Institute	S. L. Williams	7.30
2	Blue Room Engineering Society	W. D. Sprague	8.00
4	Amer. Soc. Hungarian Engrs. and Archts.	Z. DeNemeth	8.30
7–10	The American Society Mech. Engineers	Calvin W. Rice	
9	Illuminating Engineering Society	P. S. Millar	8.00
10	American Institute of Electrical Engineers	R. W. Pope	8.00
17	New York Railroad Club	H. D. Vought	8.15
21	New York Telephone Society	T. H. Lawrence	8.00
22	Municipal Engineers of City of N. Y.	C. D. Pollock	8.15

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Terms expire at Annual Meeting of 1909

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Terms expire at Annual Meeting of 1910

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Terms expire at Annual Meeting of 1909

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Terms expire at Annual Meeting of 1910

H. L. GANTT ..... New York
I. E. MOULTROP ..... Boston, Mass.
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Terms expire at Annual Meeting of 1911

TREASURER

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SECRETARY

CALVIN W. RICE ..... 29 West 39th Street, New York

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CHAS. B. DUDLEY (4)

NOTE.—Numbers in parentheses indicate length of term in years that the member has yet to serve.

SPECIAL COMMITTEES

1909

On a Standard Tonnage Basis for Refrigeration

D. S. Jacobus — G. T. Voorhees
A. P. Trautwein — Philip De C. Ball
E. F. Miller

On Society History

John E. Sweet — H. H. Suplee
Chas. Wallace Hunt

On Constitution and By-Laws

Chas. Wallace Hunt, *Chairman* — F. R. Hutton
G. M. Basford — D. S. Jacobus
Jesse M. Smith

On Conservation of Natural Resources

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Charles Whiting Baker — M. L. Holman
Calvin W. Rice

On International Standard for Pipe Threads

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On Thurston Memorial

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On Standards for Involute Gears

Wilfred Lewis, *Chairman* — E. R. Fellows
Hugo Bilgram — C. R. Gabriel
Gaetano Lanza

On Power Tests

D. S. Jacobus, *Chairman* — L. P. Breckenridge — Edward F. Miller
Edward T. Adams — William Kent — Arthur West
George H. Barrus — Charles E. Lucke — Albert C. Wood

On Land and Building Fund

Fred J. Miller, *Chairman* — James M. Dodge
R. C. McKinney

SOCIETY REPRESENTATIVES

1909

On John Fritz Medal

HENRY R. TOWNE (1) — F. R. HUTTON (3)
AMBROSE SWASEY (2) — CHAS. WALLACE HUNT (4)

On Board of Trustees United Engineering Societies Building

CHAS. WALLACE HUNT (1) — F. R. HUTTON (2)
FRED J. MILLER (3)

On Library Conference Committee

J. W. LIEB, JR., CHAIRMAN OF THE LIBRARY COMMITTEE OF THE AM. SOC. M. E.

On National Fire Protection Association

JOHN R. FREEMAN — IRA H. WOOLSON

On Joint Committee on Engineering Education

ALEX. C. HUMPHREYS — F. W. TAYLOR

On Government Advisory Board on Fuels and Structural Materials

GEO. H. BARRUS — P. W. GATES
W. F. M. GOSS

On Advisory Board National Conservation Commission

GEO. F. SWAIN — JOHN R. FREEMAN
CHAS. T. MAIN

On Council of American Association for the Advancement of Science

ALEX. C. HUMPHREYS — FRED J. MILLER

NOTE.—Numbers in parentheses indicate length of term in years that the member has yet to serve.

OFFICERS OF THE GAS POWER SECTION

1909

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L. V. Goebbels — S. A. Reeve
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Louis C. Doelling

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STUDENT BRANCH	AUTHORIZED BY COUNCIL	HONORARY CHAIRMAN	PRESIDENT	SECRETARY
	1908			
Stevens Inst.of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter		C. F. Hirshfeld
	1909			
Armour Inst. of Tech. Chicago, Ill.	March 9	C. F. Gebhardt	N. J. Boughton	M. C. Shedd
Leland Stanford, Jr., University, Palo Alto, Cal.	March 9	W. F. Durand	P. H. Van Etten	H. L. Hess
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianella
State Agri. College of Oregon, Corvallis, Ore.	March 9	Thos. M. Gardner	J. J. Karstetter	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. A. Kirk	J. R. Jackson
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	H. S. Coleman	John Garver
New York Univ., New York.		C. E. Houghton		Andrew Hamilton
Univ. of Illinois, Urbana, Ill.		W. F. M. Goss	W. F. Colman	S. G. Wood
Penna. State College State College, Pa.				